In

# EXPERIMENTAL RESEARCHES

IN

# STEAM ENGINEERING,

BY


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MADE, PRINCIPALLY, TO AID IN ASCERTAINING THE COMPARATIVE ECONOMIC EFFICIENCY OF STEAM USED WITH DIFFERENT MEASURES OF EXPANSION, AND THE ABSOLUTE COST OF THE POWER OBTAINED THEREFROM IN WEIGHTS OF FUEL AND STEAM: THE CAUSES AND QUANTITIES OF THE CONDENSATIONS IN THE CYLINDER: THE ECONOMIC EFFECT OF STEAM JACKETING, AND STEAM SUPERHEATING, AND OF VARIOUS PROPORTIONS OF CYLINDER CAPACITY FOR THE SAME WEIGHT OF STEAM USED PER STROKE OF PISTON: THE ECONOMIC AND ABSOLUTE EVAPORATIVE EFFICIENCIES OF BOILERS OF DIFFERENT TYPES AND PROPORTIONS: THE COMPARATIVE CALORIFIC VALUES OF DIFFERENT COALS AS STEAM GENERATORS: THE PERFORMANCES OF UNITED STATES WAR SCREW STEAMERS, &C., &C., &C.

**THE WHOLE BEING ORIGINAL MATTER COMPOSED OF EXTENSIVE EXPERIMENTS MADE BY THE U. S. NAVY DEPARTMENT.**

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# LIST OF PLATES.



* NOTE.—The numbering on these Plates is incorrect. Plate III should be IV, and Plate IV should be III.

# PREFACE.

WITH two exceptions, the various papers forming this volume are relations of experiments in steam engineering, made at the instance of the writer and, in many cases, under his own supervision; when, however, conducted by others, they were carried on in accordance with minutely detailed directions furnished by him; and in all cases, he has made the calculations and inferences, and written the narrations. These facts are mentioned merely to determine the authority and to place the responsibility upon the proper person. That only for which he is not responsible, is the accuracy of the measured quantities composing the data in the experiments not personally conducted by him. For these he was compelled to rely on the care and sagacity of the engineers who acted under his instructions. The two exceptions mentioned, are the papers on the United States Screw Frigates "MERRIMACK," "WABASH," "MINNESOTA," "ROANOKE," and "COLORADO," of 1854; and on the United States Screw Sloop "BROOKLYN." These contain the precise dimensions and weights of the vessels and of their machinery; together with an exact determination of their normal performance, both for the maximum and the mean, as deduced from a laborious collation of the whole of their steam logs and indicator diagrams; for the accuracy of which the engineers of the vessel are responsible, the writer being answerable for only the truth of his deductions. These papers may, in effect, be considered as the narration of the data and results of an experiment on a grand scale, made with the hulls and machinery of these steamships.

From the great paucity of reliable practical data on the subjects of the experiments recorded in this volume, their value can scarcely be overrated. Let them be compared, either in magnitude, novelty, or completeness, with what has previously been published, and the reader will quickly appreciate the difference. The cost of such experiments alone, must, in most cases, be an insurmountable bar to their being made by individuals; and it is only governments or large corporations that can afford to undertake them. These are generally under the control of persons who have but little knowledge of, and take less interest in, engineering problems. They are content with getting along about as well as others, and are averse to the trouble and cost of making investigations whose immediate use is not apparent and commercially valuable to them. The great liberality of the Navy Department in this respect is well known, and the engineering community is indebted to it for the practical determination of many facts of the highest interest, which, but for that, had remained unknown.

The engineering periodicals and newspapers of the day will indeed be found teeming with results of the *mirabile dictu* class from experiments on new inventions or modifications of machinery, or old ones revived. In these cases, however, the results alone are given, and but rarely any attempt at statements of the details and processes by which they were obtained, and when such are offered they are found to be so meagre and contradictory as to be worthless. These results are generally from patented contrivances and are published by the patentees, or they are slight modifications of well-known machinery and are published by the builders.

In both cases the publication is simply of the nature of an advertisement, and deserves little or no consideration. The purpose of the inventor and the builder is solely to promote their private interest, and their statements are framed with a view to that alone. No experiments are entitled to confidence and consideration save those made by competent and disinterested persons where there can be no private interests to be subserved.

It will be perceived that this volume is a collection of original engineering statistics with the general laws deduced from them. Science is nothing but a similar collection of statistics, and all the real knowledge we possess is derived from observation and comparison. As, therefore, sagacious comparison of the results of accurate observation is our sole teacher, the first care of the investigator must be to acquire precise experimental data from which to proceed, instead of commencing with imaginary assumptions that can only end in a vain parade of misdirected skill. The statistics of engineering should be examined in two ways: first, inductively for the single case in question; and, then, by grouping the results from a large number of similar cases, we should advance to the broad generalization applicable in practice. For although a single "positive instance" in pure science may be all that is required, yet in the complicated matter of physical effects, which are the joint production of many natural causes and are influenced by a variety of circumstances, we must be cautious to generalize not from single facts but from groups of facts.

A fact, to be of practical authority in engineering, must be derived from experiments made on the scale and under the conditions of actual practice, both as regards the magnitude of the apparatus and the time of continuance; they must be many too and repeated under every variety of circumstance. Short experiments or on too small a scale are uncertain; so likewise, are incomplete experiments, or those in which every fact connected with them is not ascertained and properly recorded, in order that checks may be furnished, and that the point at issue may be eliminated of all accidental circumstances and presented as an independent truth. To establish a physical fact we must not confine ourselves to one mode of investigation, we must employ various methods, and not accept it until we find that all, when properly cleared of extraneous influences, conduct to the same result.

In the descriptions of the experiments recorded in this volume, there will doubtless be found much of repetition and, also, statements of minutia of attending conditions which may be considered unnecessary. Notwithstanding this objection, which might have been avoided by the easy effort of omission, it is believed that no apology is required for their insertion. The writer, in his own experience, has frequently been disappointed by finding only such parts of an experiment related which the author's own theoretical views made important, to the exclusion of other parts most essential in answering a variety of different questions which would first occur to other minds. It is concluded, therefore, that minuteness and completeness are desirable although at the expense of prolixity; and that, in subjects of this nature, conciseness and condensation should not be sought by the omission of even the most trivial fact. Again, as regards repetition, if the governing conditions in a course of experiments are occasionally briefly summed,

the memory being less fatigued, their sequences are more easily conceived by the mind, and if the book is thereby lengthened, the reader's time is still more shortened.

The preparation of this volume for the press has been performed in addition to the regular business of the writer. It has been done after the labors of a long day; and is the product of a weary hand and an attention overtasked by a great number of pressing and important duties whose requirements had to be immediately satisfied, and in which responsibility could neither be avoided nor divided. These remarks are offered in apology for the errors that may have crept into the calculations, if any such be found; and for the faults of style, arrangement, or mode of treatment. The latter, too, are of but little consequence when the few readers are considered which such a work must have from its very nature; and that those few will be solicitous only for the truths contained without attaching much importance to the manner in which they are conveyed.

The manuscript of these pages was offered to several publishers who declined to print it because they considered such a work too unremunerative. The Navy Department had no funds that could be appropriated to that purpose, but granted permission for its publication to several eminent engineers who defrayed the cost in order to have the information in a convenient form.

A large portion of the volume is devoted to experiments made to ascertain by practical results the relative economy of using steam with different measures of expansion. These results are so opposed to the popular belief in the great economic gain to be obtained from the use of steam with high measures of expansion, according to the hypothetical law of MARIOTTE which has been so long an undisputed article in the creed of engineering, that a reformer exposes himself to the usual fate given by the worshippers of an *idolon fori* to those who attempt its overthrow. Nevertheless, when the subject is properly examined, subject to even the erroneous assumptions of the law considered as the expression of a physical truth, it will be seen that the fallacy of this expectation can be demonstrated by a plain application to the case of a steam engine. Such an application, made by simple arithmetic and level to the meanest capacity, will be found in the following Table, in which are given the data and calculated results showing the relative *theoretical* economy in rapport of fuel of using steam of 40 pounds per square inch total maximum pressure in a condensing engine, with different measures of expansion under normal conditions. These conditions are as follows, namely:—1st. The same initial pressure in the cylinder. Now as this pressure is determined by purely boiler considerations, it is obvious that whatever pressure can be used with one measure of expansion can be used with all, the capacity of the cylinder being properly proportioned to the work to admit this equality. 2d. The including of the effect of the space in the cylinder clearance and steam passage. This space is, for the same type of engine working with the same stroke and speed of piston, sensibly proportional to the space displacement of the piston per stroke, irrespective of the mean steam pressure in the cylinder. 3d. The including of the effect of the back pressure against the piston during the stroke, and of the pressure required to work the engine, including shafting, *per se*. For the same type of engine this back pressure is constant. The pressure required to work the engine, including shafting, *per se*, is nearly constant for cylinders of medium capacity and over: *ceteris paribus*, smaller cylinders must require a somewhat greater pressure per square inch of piston to work them than larger ones. After deducting from the total average pressure upon the piston during its stroke, the back pressure and the pressure required to work the engines *per se*, the remainder is the pressure which equilibrates the load and the friction of the load. Now as the friction of the load is a constant co-efficient of the load, it may, in all calculations

of comparative results, be considered as a part of the load without affecting the conclusions. 4th. That the engine develops equal net power in equal time with equal stroke and speed of piston. This is obviously a correct condition; for the practical problem is the doing of a given amount of work in a given time with a given speed of movement, and whatever length of stroke of piston is practicable with steam used with one measure of expansion is practicable for all other measures. The net power is, of course, the power due to the pressure which remains upon the piston after deducting the sum of the back and friction pressures. Now in order that the engine with the same initial pressure in the cylinder may develop the same net power in the same time with the same speed and stroke of piston, the area of that piston must be made inversely proportional to the net pressure, that is to the pressure which remains after deducting from the total average pressure upon the piston the sum of the back and friction pressures. Hence, as with higher ratios of expansion of the steam, the average piston pressure with equal initial pressure is lower, larger engines must be employed with the higher measures of expansion for the development of the power. 5th. That there occurs in the cylinder a condensation of so much of the steam as is required to furnish the heat transmuted into the total power developed by the engine. That this condensation proportionally reduces the pressure in the cylinder; and that the resulting water of condensation, precipitated upon the interior surfaces of the cylinder, is re-evaporated at the expense of the fuel during the time the exhaust is open and the condenser is in communication with the cylinder.

These conditions, it will be observed, greatly affect the economic result from using steam expansively, though supposing it to expand according to the theoretical laws of MARIOTTE and GAY LUSSAC, modified only by the reductions due to the transmutation of a portion of its heat into the power of the engine, according to JOULE'S equivalent. They suppose the steam to lose no heat by external radiation, nor by its expansion *per se*, and to require additionally from the boiler only heat enough to re-evaporate the water resulting from the condensation to produce the power. The economic result thus obtained is obviously the highest theoretically possible; and very greatly higher than can be practically realized; for it has been indisputably determined by many experiments, made by different persons under widely varying circumstances, that the condensation in the cylinder additional to that which is due to the production of the power, rapidly increases with the measure of expansion. For example:—this additional condensation, ascertained at the end of the stroke of the piston, is, for a mean, about 33 per centum of the weight of steam entering the cylinder when it is cut off at one-fifth the stroke of the piston from the commencement; about 20 per centum when cut-off at one-third the stroke; about 14 per centum when cut off at one-half the stroke; and about 6 per centum when cut off at two-thirds the stroke. These quantities, being obtained by comparing the weight of steam used by indicator with the weight by tank measurement, cannot be considered as expressing the whole additional condensation which took place in the cylinder during the stroke of the piston, but only the condensation present at the moment the stroke of the piston was completed: there might have been, and doubtless was, more condensation, a portion of which having been re-evaporated during the latter part of the steam stroke under the lessening pressure caused by the expansion, re-appeared as steam at the end of the stroke of the piston, and decreased the apparent amount of condensation as measured at that point.

In the following table, in which the lines have been numbered for convenience of reference, the calculations have been made for the comparative theoretical economic results with the steam cut off in the cylinder at $\frac{1}{12}$th, $\frac{1}{6}$th, $\frac{1}{4}$th, $\frac{1}{3}$d, $\frac{5}{12}$ths, $\frac{1}{2}$, $\frac{7}{12}$ths, $\frac{2}{3}$ds, $\frac{3}{4}$ths, $\frac{5}{6}$ths, and $\frac{11}{12}$ths, and for full stroke. The space comprised in the clearance and steam passage at one end of the cylinder being taken at one-twelfth the

space displacement of the piston per stroke, the measures of expansion for the steam in the different columns under the above headings are those given in line 1.

Line 2 contains the pressure of the steam in the cylinder above zero before it was cut off. It has been taken at 40 pounds per square inch, or about 25 pounds above the atmosphere, as about the maximum used with condensing engines.

Line 3 contains the mean pressure of the steam in pounds per square inch above zero during the stroke of the piston, calculated according to the theoretical laws of MARIOTTE and GAY LUSSAC, and inclusive of the effect of the steam in the clearance and steam passage.

Line 4 contains the per centum of the steam that had entered the cylinder before being cut off, which is condensed in the cylinder to furnish the heat transmuted into the total power developed by the engine. These quantities are calculated according to JOULE'S equivalent, and supposing the steam to produce its entire theoretical economy according to the laws of MARIOTTE and GAY LUSSAC. The condensation of this portion of the steam, which is taking place continuously throughout the entire stroke of the piston, proportionally reduces the steam pressure on line 3.

Line 5 contains the true thoretical mean pressure of the steam in pounds per square inch above zero during the stroke of the piston. These quantities are those on line 3 reduced by the sum of the per centum on line 4. The theoretical mean pressure cannot be greater than here given and may be found, when direct experiments on the expansion of steam are made, very much less.

Line 6 contains the sum of the back and friction pressures in pounds per square inch of piston. By the back pressure is meant the uncondensed vapor in the cylinder opposing the movement of the piston, and it is taken at 3 pounds per square inch, as about the minimum in practice. By the friction pressure is meant the pressure required to work the engine and shafting, *per se*, and it is taken at 2 pounds per square inch, as about the average in practice.

Line 7 contains the net effective pressure in pounds per square inch. It is the pressure utilized, or that can be applied to useful work; and is the remainder of the quantities on line 5 after deducting those on line 6.

Line 8 exhibits the relative areas of cylinders for developing with the pressures on line 7, equal net power in equal time with equal speed and stroke of piston. These quantities also represent the relative capacities of such cylinders, because the stroke, by supposition, is constant. They are the quantities on line 7 expressed proportionally.

Line 9 contains the relative weights of steam used for developing equal net powers in equal time with equal speed and stroke of piston. They are obtained by multiplying the quantities on line 8 by the sum of the decimal fraction expressing the fraction of the stroke of the piston completed when the steam is cut off, and the decimal fraction expressing the fraction of the stroke of the piston developing a space equivalent to the space in the clearance and steam passage at one end of the cylinder. These weights do not include the steam required to furnish heat in the cylinder for re-evaporating the water of condensation resulting from the production of the power, (line 4.)

Line 10 contains the same quantities as line 4, and represents in per centum of the quantities on line 9, the additional fuel required in the furnace to re-evaporate the water (line 4) due to the condensation of steam in the cylinder to furnish the heat transmuted into the power of the engine.

Line 11 contains the relative weights of fuel used for developing equal net power in equal time with equal speed and stroke of piston. These quantities are those on line 9 increased by the per centum on line 10.

Line 12 shows the per centum of theoretical gain or loss due to the use of steam with the different measures of expansion, and under the given conditions; assuming the cost of the net power when cutting off at

two-thirds of the stroke of the piston from the commencement for unity. The gains are indicated by the sign (+) plus, and the losses by the sign (—) minus.

The quantities in this table, though perhaps not numerically rigorously exact, are sufficiently so to give precision to our ideas, for their variation from strict accuracy under the assumptions will be found but very small. Of course they apply, in degree, only to the particular case selected of an initial steam pressure of 40 pounds per square inch above zero in the cylinder, with the sum of the back and friction pressures equal to 5 pounds per square inch. With greater initial pressures, *ceteris paribus*, the economic results would be more favorable to the high measures of expansion; as would also be the case if the sum of the back and friction pressures could be reduced. And, *vice versa*, with less initial pressures, or with the sum of the back and friction pressures greater, the economic results would be more favorable to the low measures of expansion. It is interesting to know, however, that with the pressures actually employed in the best practice with condensing engines, and with the engine properly proportioned in size to its work, the commercial theoretical value in fuel of using the steam at the most advantageous point of cutting off (one-fourth of the stroke of the piston from the commencement) is only $17\frac{3}{4}$ per centum more than when cutting it off at two-thirds of the stroke of the piston from the commencement. That is to say, using the steam with an expansion of four times is *theoretically* only $17\frac{3}{4}$ per centum more economical than using it with an expansion of one-and-a-half time. *Practically*, there must be made from this $17\frac{3}{4}$ per centum the very serious reductions due to the well known and considerably greater condensations in the cylinder, additional to that included in the table, when using the steam with the higher measure of expansion, leaving it doubtful whether gain or loss will be practically realized by cutting off shorter than about two-thirds of the stroke of the piston from the commencement, and making it certain that the difference upon either side will be practically insignificant. When cutting off shorter than one-fourth of the stroke of the piston, the theoretical gain rapidly declines until, when cutting off at one-twelfth of the stroke, it becomes only $1\frac{7}{8}$ per centum of the cost when cutting off at two-thirds.

By observing the quantities in the table it will be seen that the "true theoretical mean pressures of the steam above zero during the stroke of the piston" (line 5) is less than that due to the hyperbolic curve developed according to the MARIOTTE law alone, whereas these pressures, as obtained from engines in actual practice by means of the indicator, generally show more. The expansion curve as given by the indicator shows too much mean pressure for the MARIOTTE law if referred to the pressure at the point of cutting off, and too little mean pressure if referred to the pressure at the end of the stroke of the piston; its character also varies for every point of cutting off, departing more and more from the true curve as higher and higher measures of expansion are used. It may then be asked why, if the true theoretical pressure should be less than that due to the MARIOTTE law alone, is the practical pressure greater? The answer is that the additional pressure observed during the last portions of the stroke of the piston is supplied by the re-evaporation of the water resulting from the condensation of the steam in the cylinder during the first portions of the stroke of the piston (and sometimes to the leaking in of air at the cylinder-head joints and stuffing-boxes); this re-evaporation taking place under the lessened pressure due to the expansion of the steam, and mainly at the expense of the heat imparted to the metal of the cylinder by the steam of boiler pressure during the first portions of the stroke, which abstraction of heat must be re-supplied by the next charge of boiler steam and at the expense of the fuel in the furnace. The enormous amount of condensation in the cylinder may be inferred from the fact that, when cutting off at about one-fifth of the stroke of the piston from the commencement, and notwithstanding the time and great reduction of pressure given for the re-evaporation,

**TABLE** SHOWING THE RELATIVESSURE IN THE CYLINDER WITH DIFFERENT MEASURES OF EXPEQUAL TIME WITH EQUAL STROKE AND SPEED OF PISTON: THA'E HEAT TRANSMUTED INTO THE TOTAL POWER DEVELOPED, IN DER EXCEPT THAT WHICH IS DUE TO THE TRANSMUTATION OF THE TIME THE CYLINDER IS IN COMMUNICATION WITH THE CO IS 40 POUNDS PER SQUARE INCH, WITH THE SUM OF THE BACK HE CLEARANCE AND STEAM PASSAGE AT ONE EIROKE.

| No. of Line. | | STEAM IS CUT OFF. | | |
|---|---|---|---|---|
| | | 5/6 | 11/12 | 1 |
| 1 | Space occupied before expansion being 1, the space occ be:— . . . . . | 1⅕ | 1 1/11 | 1 |
| 2 | Initial Pressure of the steam above zero, in pounds per s | 40 | 40 | 40 |
| 3 | Mean Pressure of the steam above zero during the strok OTTE's and GAY LUSSAC's laws, in pounds per squai | 39·42 | 39·870 | 40·000 |
| 4 | Per centum of the steam entering the Cylinder condensed muted into the Total Power of the Engine, | 6·67 | 6·18 | 5·72 |
| 5 | True theoretical Mean Pressure of the steam above zer Piston in pounds per square inch, . | 36·79 | 37·40 | 37·71 |
| 6 | Sum of the Back and Friction Pressures, in pounds per s | 5·00 | 5·00 | 5·00 |
| 7 | Net Effective Pressure in pounds per square inch, | 31·79 | 32·40 | 32·71 |
| 8 | Relative Areas of Cylinder for Developing Equal Net P Equal Speed and Stroke of Piston, . | 1·029 | 1·009 | 1·000 |
| 9 | Relative Weights of Steam used for Developing Equal N with Equal Speed and Stroke of Piston, . | 0·943 | 1·009 | 1·083 |
| 10 | Additional Fuel required to Re-evaporate the Water d Steam in the Cylinder to furnish the Heat Transmu Engine, in per centum of the weight of Steam ent duce the pressures, . . . | 6·67 | 6·18 | 5·72 |
| 11 | Relative Weights of Fuel used for Developing Equal Net Equal Speed and Stroke of Piston, . | 1·006 | 1·071 | 1·145 |
| 12 | Per centum of Theoretical Economic Gain or Loss due Different Measures of Expansion, under the above Cost when Cutting off at two-thirds the Stroke of mencement for unity, . . . | — 12·53 | — 19·80 | — 28·08 |

the difference between the weight of steam which entered the cylinder by tank measurement, and the weight present at the end of the stroke of the piston by indicator measurement, is about 33 per centum of the former. This large proportion is, of course, re-evaporated and passed over into the condenser during the exhaust stroke of the piston, that is, during the time the cylinder is in communication with the condenser. Were it not for the pressure supplied by the re-evaporation, the expansion curve would be nearly a straight line, and would fall rapidly from the point of cutting off to the back pressure line, so that the area of the indicator diagram after the closing of the expansion valve would be extremely small compared with the area as given by the indicator. Now, as the pressure supplied by the re-evaporation is at the expense of the fuel in the furnace and almost as much so as if drawn directly from the boiler, it follows that what appears as expanded steam on the indicator diagram is really mostly unexpanded steam; steam boiled off in the cylinder instead of in the boiler, but at the expense of the fuel in the furnace transferred to the cylinder by the boiler steam. The curved line of the steam pressures after the closing of the expansion valve would naturally result from the continuously lessening velocity of the piston as it approached the end of its stroke, and the continuously lessening pressures, both of which make more favorable conditions for the re-evaporation as the end of the stroke is approached, and cause the pressure to be there proportionally higher. Now it is obvious that the steam boiled off in the cylinder after the closing of the expansion valve, and at the expense of the fuel in the furnace, must lose its character of expanded steam, and be obtained at nearly the same cost of fuel as if drawn direct from the boiler. I say nearly the same cost, for the movement of the piston, even in that case, must produce some expansive action. Hence it will be understood how delusive are all expectations of the economy of using steam with high measures of expansion, founded on the evidence either of indicator diagrams or of the MARIOTTE law, the former showing merely the actual steam pressures on the piston, but without giving any clue to the cost of fuel producing them, or to the fact of whether they are the result of expansion or of re-evaporation, or of air leakage when the steam is expanded below the atmospheric pressure. The enormous condensation in the cylinder resulting from the use of steam expansively, and the increased *pro rata* cost of the back and friction pressures when the initial pressure is constant, restrict the practical economy to the very low measures of expansion, ranging, according to size and type of engine, from one-and-a-half to twice, found by experiment to be as economical as any in fuel; and, from the lessened size of cylinder for doing the same work, greatly more economical in first cost and after repairs.

Among the causes of the great condensation observed in the cylinders of steam engines when the steam is used very expansively, may be named the condensation due to that expansion *per se;* that is, independent of and in addition to the production of the power. On this subject there may be briefly remarked that, mechanical power, under whatever conditions it may be developed, is the result of a change of state in some body somewhere. This change of state is an effect, and cannot be produced without the action of an efficient cause. In all that relates to steam, so far as our present knowledge extends, this cause is heat. When steam is used without expansion the body undergoing change of state is the water in the boiler, which is converted from the liquid into the vaporous state, and at the expense of the heat of the furnace. When steam is used expansively there is also a change of state, and of the same kind, namely, an enlargement of bulk—a change from one degree of density to a less—and this change, like the former, must also be at the expense of heat somewhere; and as the communication with the boiler is shut off, the necessary heat must be obtained from the steam itself. But the abstraction of heat from steam condenses it, and thereby takes it from the condition of power which it can exert only while it retains the vaporous form. The economic gain due to the expansion, then, is the difference between the dynamic effect produced by the expansion,

and its cost in heat derived from the steam, compared with the dynamic effect produced by the expenditure of an equal amount of heat derived from the fuel in the boiler furnace.

According to the well established laws of thermodynamics, steam, in overcoming any force of any kind, must suffer condensation. For instance, in rising in vacuo in a vertical column against gravity, exclusive of overcoming any resistance except that due to its own weight, it will condense. It will condense in transporting itself from the boiler to the cylinder, and from the cylinder to the condenser; and in following the piston down the cylinder, supposing the load to be carried by an external power. In fact, it cannot produce movement in its own particles of any kind without expenditure of mechanical force and consequently transmutation of heat and resulting condensation.

Starting from the indisputable fact that steam, in overcoming any force of any kind, must suffer condensation, it only remains to show that if the particles of steam have an attraction for each other, that attraction is a force which must be overcome in the process of expansion *per se*, and will involve the transmutation of its equivalent of heat which can be obtained only from the steam itself furnished by the condensation of a portion of it. Now the particles of steam, like those of all other matter, have a strong attraction for each other, shown by their instantly coalescing on the removal of the repelling agency of the heat. The mechanical power expended in the expansion *per se* of steam is, therefore, due to and measured by two causes: first, overcoming the attraction of the particles for each other, and then overcoming their inertia involved in their removal to greater distances apart. Until the particles of steam can be shown to have neither attraction for each other, nor weight, they cannot be removed farther apart without the expenditure of mechanical power; and as this mechanical power can only be developed at the expense of the heat in the steam, its annihilation must be accompanied by condensation. This is the argument for the condensation of steam *per se*, irrespective of external work done, and may be considered as due to the internal work done on its own particles.

That steam does not follow the law of perfect elasticity is proven by the late and direct experiments of Fairbairn and Tate, and was long since suspected to be the case. They found the volumes to be less proportionally than the pressures, which is accounted for by Berzelius on the supposition that the closer the molecules are brought together the stronger becomes their attraction of cohesion which thus neutralizes a part of the separating power of the heat. Now, whether a vapor or gas follows the laws of Mariotte and Gay Lussac, depends entirely on the force of attraction of its molecules relatively to the repulsive energy of the heat. In proportion as the former is stronger the bulks to which the vapor or gas may be expanded will not be in the inverse ratio of the pressures, but in a less ratio. And in proportion as the latter is stronger the bulks to which the vapor or gas may be expanded will not be in the inverse ratio of the pressure, but in a greater ratio. Hydrogen gas is an example of the latter case, and carbonic acid gas of the former, according to the late experiments of Regnault. It is probable that every gas and vapor has not only a ratio of expansion peculiar to itself, but that this ratio, for the same gas, varies at different pressures, and there may be some gases which exactly agree with the hypothetical law: atmospheric air seems to follow it closely. Now the condensibility of a given vapor is the measure of the force of the attraction of its particles for each other, and steam being the most easily condensible vapor of which we have knowledge, it follows that its bulks after expansion should be in a far less ratio to the pressures than other vapors and gases. Direct experiments on this point are wanting: they are a great desideratum, but from the difficulty of making them, it will have for the present to be substantiated by indirect ones.

The fact of the condensation of steam as a consequence of its free expansion, that is of its expanding without performing work, has been rendered certain by some late experiments of Dr. JOULE and Professor W. THOMSON on the thermal effects of the free expansion of various elastic fluids, the results of which were communicated by the experimenters to the "British Association for the Advancement of Science," and printed in its transactions for 1861. Earlier experiments by Dr. JOULE led him too hastily to conclude that if the expansion of a gas is managed in such manner that its dilatation performs no external work, no thermal effect is produced, the temperature of the gas after the expansion remaining exactly the same as before the expansion. The accuracy of these experiments, however, had been doubted by Professor THOMSON, who showed that such a result could be true only of a perfect or hypothetical gas, and that the correct experimental result would differ from it in proportion as the real gas differed from the hypothetical one. Now a perfect gas is an imaginary fluid composed wholly of centres of repulsion. It is destitute of weight and cohesion. Its parts have no attraction for each other. It is perfectly elastic, has no latent heat, and is a perfect medium for sensible heat which, as the gas has no corpuscular constitution, produces no internal work upon it during any change of bulk. It is obvious that the free expansion of such a fluid could not, from its very definition, be attended by any thermal effect. There is no latent heat to be increased during the expansion, which increase is at the expense of the sensible heat, and is the measure of the internal work done on the molecules of a physical gas; and no external work done which would have required the transmutation of a portion of the heat into mechanical power; consequently the temperature of the hypothetical gas must remain unaltered by the fact of the expansion.

The plan of experimenting devised by Professor THOMSON, and carried on by him in concert with Dr. JOULE, in order to show the certain thermal effect produced upon the expansion of elastic fluids without performing external work, consists in employing a porous plug through which the fluids, confined under a high pressure, escape. The plug requires to be "of considerable thickness; for if a thin one is used, there will be a rapid conduction of heat from the high to the low pressure side, and also an irregular effect arising from the action of numerous jets of air instead of a tranquil flow to the low pressure side. Hence they found a too large cooling effect when a diaphragm of leather was used, in which case even hydrogen showed a slight cooling." "Their first experiments on a very small scale and with a very imperfect apparatus, decisively exhibited a lowering of the temperature of air on passing the plug; thus showing a non-observance of exact gaseous law."

"The phenomena of a jet of air are highly interesting. Issuing at a high velocity from a vessel in which it is confined at high pressure, its actual temperature may readily be made 200° below the zero of Fahrenheit. But this very low temperature cannot be easily exhibited, because if a thermometer is immersed in the jet the friction of air gives rise to heat which nearly neutralizes the cold. The temperature of one part of a jet may thus be hundreds of degrees different from that of another part. The authors have, in fact, shown that a thermometer may be so placed in a jet as to experience either cold or extreme heat. Hence the absolute necessity in their experiments of a porous plug, which will allow the air to issue in a tranquil flow without jets or rapids."

"A general result they have arrived at in transmitting elastic fluids through a porous plug is, that the thermal effect is proportional to the difference of pressures on the opposite sides."

"A diminution of temperature takes place in all the gases tried except hydrogen; and this diminution or cooling effect is decreased when the temperature is raised, in such sort as to make it certain that at 300°

or 400° it would vanish altogether and be followed by a heating effect, as is observed in hydrogen at low temperatures."

"In different gases the cooling effect is very various. It is five times as great in carbonic acid as in atmospheric air at low temperatures, and four times as great at the boiling temperature."

"A very remarkable fact which has been elicited by these experiments, is that a gas mixed with another does not exhibit the same thermal effects as it does when undiluted. In general a mixture of gases gives a smaller cooling effect than would be deduced from the cooling effects of the constituents. This has been verified in the dilutions of carbonic acid and hydrogen and in atmospheric air, of which each of the component gases has a larger cooling effect than itself."

The decrease in the cooling effect due to the free expansion of the gases, observed when they were highly superheated, was probably caused by the difference between their specific heat before and after expansion.

It will be noticed that in those gases which most nearly approached the condition of condensible vapors, as carbonic acid, the cooling effect produced by their free expansion was the greatest, and we are thence authorized to infer that in the case of steam—the most easily condensible of vapors—this effect would be very much increased. For we are not entitled to argue from the results of the experiments which have been made on the free expansion of the permanent gases, that the same would be obtained in degree from the free expansion of condensible vapors. They are, in this respect, entirely inapplicable, because in the former case, no lowering of temperature which has yet been produced by artificial means has effected any liquefaction of the gases; while in the latter case the slightest reduction of temperature is attended by a liquid precipitation. In the one case there is no change of condition—the substance remaining gaseous throughout:—in the other case there is a radical change in the condition of the matter, and from the vaporous to the liquid form. In the case of the gases, then, while the reduction of temperature which has been observed during the act of free expansion, is in part restored by the heat generated by the loss of *vis viva* and the mutual attrition of the particles when the act of expansion is completed and they have come to a state of rest; there can be no such restoration in the case of the vapor, because the portion liquefied is annihilated as vapor. What has once been condensed in the cylinder will remain liquid until re-evaporated by heat from the boiler communicated by means of the metal of the cylinder. The latent heat set free by the liquefaction goes to maintaining the remaining portion of the vapor in the gaseous form; and is required for that purpose because the latent heat of steam increases with decrease of pressure faster than the sensible heat decreases.

It may be objected that the cooling effect of the free expansion of steam, though certain, is too small to produce the enormous condensations which are practically known to be the consequence of using it with high measures of expansion in an engine; but we are to consider that the ultimate condensation in a cylinder due to a certain reduction in the heat of the steam, is not to be measured directly by the amount of this reduction; and that it may become, when combined with the conditions of the machine, the cause of a vastly greater condensation.

If the steam condensed directly by the cooling effect of its free expansion, were not deposited as dew upon the interior surfaces of the cylinder, to be re-evaporated under the lessened pressure of the condenser during the time the exhaust-valve continues open; but remained suspended among the portion retaining the vaporous form, and evacuated the cylinder with it, leaving the surfaces dry, a very much less condensation than that observed would follow, and by the amount of the cooling effect produced by the rapid evaporation of this dew.

To illustrate the extent to which temperature may be reduced by such process, there may be cited the fact, first shown by CONFIGLIARCHI, that water at common temperature can be frozen in an air-pump vacuum, if in presence of a large surface of strong sulphurous acid to absorb the aqueous vapor and thus assist the pump in removing it as fast as formed. And still more illustrative is the experiment of BOUTIGNY, who exhibited the phenomenon of water frozen in a red-hot crucible, as the effect of rapid evaporation. A little water being placed in a red-hot platina crucible assumes the spheroidal form and remains without sensibly evaporating. A few drops of liquid sulphurous acid—which is very volatile and readily mixes with water—is now added, when it instantly flashes into vapor carrying with it some of the vapor of the water thus brought to the condition of ordinary ebullition; and if, at this moment, the still red-hot crucible be adroitly inverted a little lump of ice will drop out.

If liquid carbonic acid gas under high pressure be permitted to rush out, it will fall in a snow of solid crystals, although it is a permanent gas at ordinary pressures and temperatures. If these crystals be now immersed in sulphuric ether, they will dissolve and, in dissolving, cool the ether; and if the whole be then placed in an air-pump vacuum, the rapid evaporation at this low temperature and pressure will go on so fast that temperatures of—150° Fahrenheit have been reached.

The evil effects of any condensation in the cylinder, whether due to the development of the power, to the expansion *per se* of the steam, or to external radiation, will become strikingly apparent when we consider that such condensation acts upon the cost of the power in fuel in a twofold manner. First, it decreases the mean pressure upon the piston and, consequently, lessens the power from a given weight of fuel to precisely that extent. Second, supposing the liquid precipitate of the condensation to be deposited upon the metal of the cylinder and wholly re-evaporated to the condenser during the time the cylinder is in communication with it, and under its reduced pressure, the principal part of the heat for this re-evaporation will be obtained from the metal of the cylinder, and was imparted to it by the steam during the time the cylinder was in communication with the boiler. The result of the re-evaporation thus effected is a great lowering of the temperature of the interior metal surface of the cylinder, to a certain depth depending on circumstances. Now all the heat thus abstracted from the metal of the cylinder has to be replaced by the heat from the boiler when the cylinder is again put in communication with it for the next stroke of the piston, and, consequently, the cost of the power in fuel is thereby increased by exactly this latter amount of heat. Hence, we perceive that cylinder condensation, due to any cause whatever, *counts twice against the fuel;* once in the direct reduction of the power, and again in the quantity of heat required to restore the temperature of the metal of the cylinder to that of the boiler steam.

Every practical engineer has had frequent opportunities of observing the great reduction of power—with equal generation of steam—that follows the foaming or priming of boilers. When this priming reaches a certain degree, much less than that which is evidenced by the slapping in the cylinder of the water thus transferred to it from the boiler, the injurious effect on the power is instantly and strongly marked by the lessened speed of the engine. Indicator diagrams, taken under these circumstances, show but slight, if any increase of back pressure; the more copious water lubrication of the piston can have but trifling effect; and as no concussion of water is heard in the cylinder, but little resistance can result from that cause. Whence, then, comes the unmistakeable reduction in the power? the same weight of steam continuing to be received by the engine from the boiler in the same time: evidently from the evaporation of this water in the cylinder during the time the latter is in communication with the condenser, and under its lessened

pressure; which evaporation is at the expense of the fuel in the boiler, but the steam from which produces no dynamic effect upon the piston because it is passed direct from the cylinder to the condenser as fast as generated. The heat required then, to evaporate the weight of water primed over is wholly lost so far as the production of power is concerned. It is obtained from the metal of the cylinder, which has received it from the boiler steam which, by the act of imparting it, has undergone condensation.

That the mere abstraction from the boiler of the quantity of hot water primed over to the cylinder could not affect the development of the power in the strongly marked manner it is observed to do, is obvious when it is considered that the loss of heat by blowing out one-half of the water pumped into the boiler, would, in the cases of average practice with condensing engines, cause a loss of only about one-seventh of the fuel consumed, and affect the revolutions of the engine in the small ratio of the cube root of unity to the cube root of six-sevenths.

The case of the effects of priming is instanced to illustrate how great are the evil results of evaporation performed in the cylinder.

The law of MARIOTTE merely expresses the *abstract quality* of perfect elasticity; it is not a physical law, because such is an expression for generalized physical facts. The general dogma of expansion is a mere hypothesis, it is not a theory, and it is impossible that any hypothetical explanations can add really to our knowledge. In all hypotheses some supposed event is thrust in between the phenomenon and the general law in which it is embraced. A hypothesis is no generalization of facts, and should only be admitted in physical science as a conjecture for the direction of a course of experiments. In this way it possesses great use; but if mere conjectural or hypothetical systems are accepted as physical truths, they are worse than nonsense, they are mischievous, filling the mind with incorrect notions of nature, and leading into grievous errors when followed as principles of action. Now experiment upon the compression of various gases has shown a variation on both sides of MARIOTTE's law, and more strongly marked as the condition of a vapor is approached; the law is thus overthrown as a physical truth, notwithstanding its extreme simplicity and the reasonably close approximation of the fixed gases within the limits of the experiments. Indeed mere simplicity, or uniform proportionality of result as a consequence of an hypothesis, is by no means a sure ground of induction to a law of nature. The simplicity is frequently only predicable of some abstract quality that may be imagined, but which is not possessed by real bodies. Nature begins with causes that, after a variety of combinations, produce the effects upon which our senses open; but we, from these effects can only ascend back to the causes through the slow and laborious process of experiment, and must thus seek truth, not in metaphysical abstractions, but in tracing the modifications due to the joint action of the various forces of nature producing the phenomena. The results of the experiments detailed in the following pages may be impugned because impossible according to a received hypothesis; but we should consider that this hypothesis by no means embraces the action of all the agents of nature, and that it is not philosophical to deny any phenomenon because it is inexplicable according to received theories. We should remember that the onward march of physical knowledge has been over the ruins of such theories; and as we now know many things to be true experimentally which were once rejected because they did not conform to received hypotheses, so we may conclude there are still in nature many phenomena which, when fixed by experiment, will overthrow theories now as firmly accepted: in the progress of thought it is often as important to overthrow as to establish, and the reformers of philosophy must be stern iconoclasts. Many of the most important truths in the science and art of engineering are of comparatively recent

demonstration, and are slowly and reluctantly accepted. There are those who will cherish a fallacy from habit, and others, because to them it seems so plausible that they cannot see how to reject it. The *vis inertia* of man has been but too well satisfied to receive whatever is well wrought out and plausibly presented, and he, who comes as a searcher and trier of these accepted theories, sounding their depths and shallows, and detecting their inconsistencies, renders good service to humanity.

No physical force is isolated in its action, but all the forces of nature—gravity, heat, electricity, chemical affinity, &c.,—exert their powers in the same arena and upon the same objects; and this multitude of causes, thus combining to produce a single physical phenomenon, is so great as to defy our limited capacity to trace their combinations, and assign the mode of action and value of each in the crowd of miscellaneous facts which strike the senses. It must always be borne in mind that no effect can have only one cause, neither can one cause have only one effect, and every effect is the sum of the causes producing it. Our only infallible guide to knowledge is experiment, and from the constancy of nature in all her operations we are authorized to make a general application of the results from even one experiment in which we have succeeded in clearing the subject from every accidental and unknown circumstance, and thus reduced the point at issue to elementary simplicity. When we have done this properly we may be certain the event obtained will be a faithful representative of what will happen in every similar case.

Our knowledge of the physical laws of steam are yet much too imperfect to entitle us to make a confident application of mathematics, and—though many books will be found filled with dry algebraic formulas, founded, to say the best for them, on purely gratuitous assumptions, and predicting results entirely opposed to what is realized experimentally—little has been done in determining the laws of expansion because, instead of collecting facts and tracing the proximate causes, an overstrained regard has been had for a comprehensive principle. Misled by a false ambition to grasp everything and predict *à fortiori*, truths have been missed that might otherwise have been seized. What is insisted on is the fallacy and fault of ignoring the physical conditions of a practical problem, or of sacrificing them silently to the neatness and perspicuity, and to the necessities and narrow scope of a mathematical theory.

The profession of the engineer is purely practical; he should no more accept abstract and subtle speculations in place of applying himself to ascertain downright matters of fact, than a pilot should content himself with a theoretical deduction of the laws of the ebbing and flowing of the sea instead of acquiring a knowledge of the direction and force of its tides. All sound knowledge in engineering must be observational—purely experimental—and not that sort of information which is derived from developing an hypothesis; or is reached by a scaffolding of assumptions of how nature might, would, could, or should proceed. Honest, sagacious experiment, long and frequently repeated under all the various conditions of actual practice, is the only road to true knowledge, and, in order that experiment be available to the world, it must be accurately and minutely described. If every engineer had made only one *complete* experiment and placed it on record in full detail, engineering would be far in advance of its present position. A collection of accurate experiments clearly illustrated by drawings of the apparatus employed, is not only a fecund source of useful engineering knowledge, but the only sure guide in its application. It is the sole teacher of the ultimate scientific faith of practical value, by showing the modifications which abstract theoretical laws undergo when embodied in physical conditions; and by determining the practical co-efficients indispensable in the employment of the formulas of physical science and of applied mechanics, it ascertains the true relations between practice and theory. Correct information alone is the secret of engineering success; but what is information? experimental facts and their right interpretation;—a single physical fact is worth many folios of argument. What are energy, enterprise, sagacity, education, and intellect without *facts* as material to act on? But

we must have many facts, and determined from every point of view. To judge from a single experiment would be injudicious; the leading experimental truths of engineering must be the results of a wide generalization deduced from large experience and based on a great mass of testimony. Amid all the variable and complicated phenomena attending engineering, *mean numerical values* must be the object of our search, as the expression of the physical laws that govern it, using our intelligence to eliminate these laws from the mass of adventitious circumstances by which they are encumbered, to trace their remote consequences and connexions, point out their limitations, show their practical importance, generality, and real value, thereby rendering them safe points of departure for farther advances, and useful in explaining and confirming new discoveries. They are thus reduced to a consistent and well arranged system of physical science, which, notwithstanding the high claim involved in the etymology of the word, is but a rational empiricism, that is to say, the observation of facts confirmed by the operation of the intellect. From this we understand how science can be made to explain many things which it could never have discovered.

The necessity for confining ourselves to the deductions from experiments only, will become obvious when we consider that in physical science we cannot assume as first principles the most general propositions, and thence descend successively by inference to the less general until we reach the final particular; because everything in nature presents itself to our perception in its individual state, compelling our investigations to begin at the end of her record instead of at its commencement. By sagaciously instituted experiment, correctly observed, and frequently repeated upon many individuals of the same class or similar nature; canvassing, examining, and comparing them together, noting their agreements and differences to the minutest degree, and rejecting all cases that are not in effect the same, however like they may be in appearance, intelligence, with much labor extracts a few general laws respecting their powers and properties, and their modes of action and of being acted on. By this process only, slow and difficult as it is, can we advance from one stage of inquiry to another, and, tracing back the processes of nature, arrive finally at the general truth of which we are in search. The true characteristic of scientific genius—and which has contributed most to the advancement of human knowledge—is that happy tact, so wonderfully possessed by NEWTON, which recognises general principles throughout the multitude of various and apparently discordant objects in which they are enveloped. Or, to borrow an illustration from another science, it is the nice perception that detects the ever present tema amid the infinite variations which compose the grand opera of nature.

Many of the following pages will be found occupied with the relation of experiments on the evaporative efficiency of boilers of different types and proportions. The relative excellence of boilers as economic generators of steam, is a problem so easily solved by direct experiment that it admits of less opinion than any other in steam engineering. Such experiments are so simple that they may be readily made by all, for but little skill is required to weigh coal and measure water in a tank; and the whole result is determined by dividing the one quantity into the other; that is the most economical boiler which evaporates the most water per pound of coal. In the presence of the practical fact thus ascertained all argument is mere impertinence, and no amount of reasoning on circulation of currents, or on a nearer or remoter approach to perfection of combustion, or on what ought to result from peculiar types and proportions, can prevail against it: Philosophical and chemical discussion are here superseded by the *experimentum crusis* in its most direct and elementary application. But there are other qualities of much importance to be considered, especially for marine boilers, with which the space occupied is a controlling condition. In that case, the problem becomes the finding of the most economical boiler in the least space. And

even more important than these is the easy access to every part for repairs and cleaning. In comparing boilers for marine purposes only such are practicable as satisfy the latter condition, after which those that best satisfy the previous ones are entitled to the choice. It is essential then, in comparing boilers to keep in view the space occupied for the generation of a given weight of steam in a given time, as well as the economic result, for nearly all the types can be made to give a maximum evaporation if the space occupied be unrestrained.

In making experiments on boilers, the correct method is to evaporate the water under the atmospheric pressure, measuring it in a tank previous to its admission in the boiler; and *taking particular care that the steam escape pipe be both very large, and attached to the boiler at its highest point, and as near its centre as possible;* otherwise much water in the form of spray will be entrained by, and carried off with, the steam current. Blowing off the steam through the usual safety-valve opening will give incorrect results: the rapidity of the steam current will induce priming, and large quantities of water will be thrown out. This has been the cause of many absurd results in experimenting with boilers.

Again, if the evaporation be performed under the atmospheric pressure, the chances of loss of water by leakage will be reduced to the minimum, as there will be only the pressure due to the height of the column of water within the boiler. When accurate results are desired no safeguard, however trivial, should be neglected.

It will be observed that in most of the boiler experiments recorded in the following pages, the boilers were new, and clean both on the fire and water sides of the heating surfaces; that they were thoroughly covered by felt, canvass, and lead; that the combustion was not forced in any case, but in many was restricted to a very moderate rate; and that great care was observed in the firing; the economic results, therefore, may be considered a maximum; and as the Pennsylvania Anthracites are not exceeded in steam generating power by any coal in the world, it is probable these experiments show the utmost possible economic evaporation attainable from steam boilers of the best types and proportions.

To judge of the absolute, as well as of the comparative merits of boilers, it is necessary to know the theoretical evaporative efficiency of the fuel used: that is, to know the number of pounds of water the pound of fuel will evaporate from the temperature of 212° Fahrenheit, under the atmospheric pressure of 14·7 pounds per square inch, provided the combustion with atmospheric air be chemically perfect, that no more air is admitted than what is rigorously required to oxidize the fuel, and that the products of combustion on escaping from the water have the same temperature as the air supporting the combustion.

The coal used in the experiments was the Pennsylvania Anthracite, a description and analysis of which are given on page 146. The portion of this anthracite which remains after deducting the earthy residuum, is composed of carbon, hydrogen, oxygen, nitrogen, and water mechanically present in its pores. Of these the oxygen, and so much of the hydrogen as may be united with it in the proportion in which they exist in water, render latent by their decomposition as much heat as their combustion developes, thus producing no evaporate effect. The nitrogen produces no evaporative effect, as it passes off without chemical combination. The water mechanically present in the pores of the fuel, not only produces no evaporative effect, but is injurious, as it appropriates for its vaporization the same amount of heat that it would do, were it in the boiler instead of being in the fuel. This water, therefore, should be added to the economic effect of the portion of the fuel that remains after deducting it.

Of the anthracite which is left after deducting the earthy residuum, 90 per centum is carbon, 1 per centum is nitrogen, and the remaining 9 per centum is about equally divided between the hydrogen, oxy-

gen, and water. Now it is the 90 per centum of carbon alone which, in the boiler, possesses any evaporative efficiency; for the nitrogen, oxygen, and water are worse than useless, and the hydrogen, at the best, can do no more than neutralize their injurious effects. We may, therefore, conclude that of the portion of the anthracite which remains after deducting the earthy residuum, (and which in this volume is called the "combustible") only nine-tenths produces an evaporative effect, and that this evaporative effect is what is due to pure carbon alone. The evaporative efficiency of one pound of such carbon, as appears from the considerations given on page 150, is 16,000 pounds of water heated 1° on Fahrenheit's scale, which taking the latent heat of steam of atmospheric pressure at 965·7° Fahr., is equivalent to 16·5683 pounds of water evaporated at that pressure from the temperature of 212° Fahr. Now the average proportion of earthy residuum in anthracite is one-sixth, leaving five-sixths for the "combustible," of which nine-tenths, or three-fourths of the original anthracite, is carbon, making the theoretical evaporation per pound of anthracite from the temperature of 212° Fahr., and under the atmospheric pressure 12·4263 pounds of water. This evaporation becomes for the pound of "combustible," as that term is used in this volume, 14·9115 pounds. We have here a limit which under no circumstances can be exceeded, and which under the conditions of a practical boiler can never be equalled.

In the practical boiler, this evaporation of 14·9115 pounds of water by one pound of anthracite combustible, under the atmospheric pressure and at the temperature of 212° Fahrenheit, will be reduced by the following circumstances:

1st. By the quantity of heat in the products of combustion on their escape from the boiler. This quantity will be measured by the weight and specific heat of the gases composing these products, and by the difference between their temperature and that of the atmospheric air supporting the combustion. The temperature of the products of combustion on leaving the boiler must always be above that of the steam within it, and under the most favorable circumstances this excess is never less than 30° Fahrenheit, and rarely less than from 150° to 200°, frequently rising to 400° and upwards.

2d. By the greater quantity of air that may enter the furnaces than is necessary rigorously, for the complete oxidation of the fuel. An attempt to restrict it to this quantity will, in some cases, be followed by incomplete combustion, owing to want of time and means for the thorough presentation—atom to atom—of the oxygen of the air and the constituents of the fuel. And although the total heat generated by the perfect combustion of a given weight of fuel is the same, whether just enough or more air be supplied than is rigorously necessary, and will be present in the products of combustion, yet the heat available for evaporation will not be the same, because the products of combustion leave the boiler at a much higher temperature than the air enters the furnaces. Hence, the greater the unnecessary quantity of air supplied, the greater will be the loss of heat in the escaping products of combustion. For example, with just sufficient air and a complete oxidation of the carbon, the initial temperature of the resulting gases will be 4347° Fahrenheit, and if they leave the boiler with a temperature of 500° Fahrenheit, the loss of heat in them will be 11·5 per centum. If, now, twice this quantity of atmospheric air be supplied, their initial temperature will be 2233° Fahrenheit, and if they escape at 500° Fahrenheit, as before, the loss of heat in them will be 22·3 per centum.

3d. By the losses of heat due to external radiation from the boiler; for it is obvious that no system of covering can entirely prevent the conduction of heat from the metal to the atmosphere.

4th. It is improbable, in any case, that the whole of the fuel is saturated with oxygen; some portion, however small, doubtless escapes in imperfect combustion.

**TABLE** EXHIBITING THE ] IN THE FOLLOWING PAGES; AND OF OTHERS, MARKED WITH AN ASEIR ECONOMICAL EVAPORATION; WITH THE GIVEN RATE OF COMBUSTICMBUSTIBLE PORTION OF ANTHRACITE.

| LOCATION OF BOILER. | REMARKS. |
|---|---|
| U. S. S. Mount Vernon, | Horizontal Fis 3½ inches. Extreme length 9 feet 6 inches. |
| U. S. S. Valley City, | " 's 4 inches. Extreme length 10 feet. |
| U. S. S. Crusader, | " 's 4 inches. Extreme length 12 feet 2 inches. |
| U. S. S. Young America, | " 's 6 inches. Extreme length 12 feet 6 inches. |
| U. S. S. Monitor, | " 's 2½ inches. Extreme length 10 feet 2 inches. |
| U. S. S. Satellite, * | " 's 6 inches. Extreme length 18 feet. |
| U. S. S. San Jacinto, * | " 's 3 inches. Extreme length 7 feet ½-inch. |
| " " | " "serting a ferule or ring ⅛-inch thick in both ends of tubes. |
| " " | " "serting a ferule or ring ¼-inch thick in both ends of tubes. |
| " " | " "l. Ash-pit door kept nearly closed. |
| | " "bricking off 8 inches of the width of each furnace. Calorime- ; a ferule or ring ¼-inch thick in both ends of tubes. |
| " " | " " |
| " " | " "serting a ferule or ring ⅛-inch thick in both ends of tubes. |
| " " | " "serting a ferule or ring ¼-inch thick in both ends of tubes. |
| Steamer Georgeanna, * (Baltimore and Norfolk), | " "s 3 inches. Extreme length 7 feet 6 inches. |
| U. S. S. Jacob Bell, | " "s 6 inches. Extreme length 14 feet 11 inches. |
| Machine Shop, New York Navy Yard, | " "nts made with all the tubes in use. Tubes 3 inches by 8½ feet. |
| " " " " | " "ts made with two horizontal rows of tubes stopped at both ends |
| " " " " | " "ts made with three horizontal rows of tubes stopped at both ends |
| " " " " | " "ts made with four horizontal rows of tubes stopped at both ends. |
| " " " " | " "ws of tubes of each furnace stopped at both ends. |
| " " " " | " rows of tubes of each furnace stopped at both ends. |
| " " " " | " "ws of tubes of each furnace stopped at both ends. |
| U. S. S. Dragon, * | " "es 3½ inches. Extreme length 9 feet 9 inches. |
| U. S. S. General Putnam, * | " "es 4 inches. Extreme length 13 feet 10 inches. |
| U. S. S. Michigan, | Vertical Water tubes. |
| U. S. S. Roanoke, | " " tubes. |
| U. S. S. Passaic, | " " t of tubes. |
| U. S. S. Chippewa, * | " " t of tubes. |
| U. S. S. San Jacinto, * | " " tubes. Mean of three experiments of 72 hours each. |
| " " | " " tubes. Calorimeter reduced by placing an iron bar $\frac{3}{16}$-inch of tubes, both at back and front of tube boxes. |
| " " | " " ate in front of tubes. |
| U. S. S. Wyandotte, | " " t of tubes. |
| " " | " " t of tubes. |
| U. S. S. Underwriter, | Horizontal Flue |
| U. S. S. James Adger,* | " " |
| U. S. S. Whitehead, * | Double Return I |
| Brooklyn Water-Works, | " " |
| Watermann's Experimental Engine, | Locomotive Typ Furnace, s 1¼-inch. Extreme length of tubes 3 feet. |

Let us now examine the highest economic evaporation that has been obtained with anthracite combustible from boilers of the best type and proportions; and by comparing it with what is theoretically possible, we shall be able to appreciate how near they approach perfection, and the still remaining margin for improvement.

In the accompanying Table, under the head of "Economic Evaporation" will be found the results of a considerable number of carefully conducted experiments, made on a large scale, with various boilers of different types and proportions. These results are strictly comparable. The manner of obtaining them, and all the information in connexion therewith, will be found narrated in the following pages under the names of such as are not distinguished by an asterisk (*). For those marked with an asterisk (*) the processes were exactly the same, and will be described, together with the entire detail of the boilers and experiments, in the succeeding volume.

Referring to these results, and taking the mean for the vertical water tube boilers of the U. S. Steamers "ROANOKE," "PASSAIC," "CHIPPEWA," "SAN JACINTO," and "WYANDOTTE," (first experiment with the latter) we find that with a ratio of 33·590 square feet of heating surface to one square foot of grate surface, and 7·466 square feet of grate surface to one square foot of cross area through the tubes for draught, the rates of combustion being 7·104 pounds of anthracite combustible per square foot of grate surface per hour, and 0·215 pound of anthracite combustible per square foot of heating surface per hour, there were evaporated under the atmospheric pressure, and from the temperature of 212° Fahrenheit, 13·0383 pounds of water per pound of anthracite combustible, leaving to cover all the losses enumerated under the above four kinds, only $\left(\frac{14{\cdot}9115-13{\cdot}0383\times100}{14{\cdot}9115}=\right)$ 12·56 per centum of the theoretically possible evaporation. In these cases the temperature of the escaping gases averaged about 300° Fahrenheit, or 88° above that of the steam, and supposing the combustion to have been perfect, and no more air to have entered the furnaces than was rigorously necessary, the loss due to this temperature is 6·9 per centum, leaving only 5·66 per centum to cover the loss by radiation. The average earthy residuum in these experiments was 16·06 per centum of the anthracite. The only possible improvement that can be made on this result, is the little that might be effected by reducing the temperature of the escaping gases from 88° to, say, 28° Fahrenheit above that of the steam, which would give an increase of 1⅜ per centum, but would require a considerable addition of heating surface to obtain. It may, therefore, be concluded that, with the boilers described and the rate of combustion employed, the maximum limit of economic evaporation has been obtained, and that no further improvement is possible.

When, in these boilers, the rate of combustion is increased to 10 pounds of anthracite combustible per square foot of grate per hour, equivalent to 12 pounds of anthracite, there being consumed 0·⅜ pound of combustible per square foot of heating surface per hour, the economic evaporation falls to 12·5 pounds of water per pound of anthracite combustible, making the loss $\left(\frac{14{\cdot}9115-12{\cdot}5000\times100}{14{\cdot}9115}=\right)$ 16·17 per centum of the theoretically possible evaporation: the temperature of the escaping gases rising with the increased rate of combustion, and producing the increased loss of (16·17—12·56=) 3·61 per centum by the increase of temperature.

Attention may now be given to the space occupied by the vertical water-tube boiler with the tubes arranged above the furnaces. All boilers are composed essentially of five portions. 1st, The furnace and its ash-pit. 2d, The smoke connexions. 3d, The tubes or flues. 4th, The steam room. And, 5th, The water-space. Of these, the 1st and 4th must necessarily be constant. Whatever furnace and ash-pit will answer

for one type, will answer equally well for any other. And it is obvious that the capacity of the steam room is irrespective of kind of boiler. The smoke connexions are so nearly the same in bulk for all types that the difference is immaterial. As regards the space occupied by the tubes, a little reflection will show, that, a greater heating surface can be properly packed in a given space with the vertical water tubes of two inches external diameter, than with any other arrangement of tubes or flues. And that as a necessary consequence, the water space with them will be smaller. The general cross section of this type of boiler approaches a square, the figure of all rectilineal ones having the least periphery, consequently the shell for a given area of grate and heating surface will have the minimum weight, and heat radiating surface; the less water contained will not only weigh less, but it allows steam to be more quickly raised. In general, in a given shell of boiler, and with the same furnaces and steam room, at least one-fifth more heating surface can be properly packed in a vertical water-tube boiler than in a horizontal fire-tube boiler. The water-surfaces of the vertical tubes admit more easy access for scaling, but the fire-surfaces require more time and labor to sweep off the ashes; not enough more, however, to amount to a serious practical inconvenience.

In economic evaporative efficiency, and employing the best proportions for each type, using a given shell with the same furnaces and steam room, and a combustion of 12 pounds of anthracite per hour, the vertical water-tube boiler compares as $12\frac{1}{2}$ to $11\frac{1}{3}$ for the horizontal fire-tube boiler, that is, the former is $\left(\frac{12\frac{1}{2}-11\frac{1}{3}\times 100}{11\frac{1}{3}}=\right)$ 10·3 per centum better than the latter. And as the maximum rate of combustion with natural draught is about 3 per centum less with the vertical water-tubes than with the horizontal fire-tubes, it follows that in potential evaporation, or quantity of steam producible from a given shell in a given time, irrespective of fuel economy, the vertical water-tube boiler is seven per centum superior to the horizontal fire-tube boiler.

When the rate of combustion is carried up by means of a blower to 24 pounds of anthracite per square foot of grates per hour, the economic efficiency of the vertical water-tube boiler falls from 10·3 to 3·9 per centum better than that of the horizontal fire-tube boiler, its potential evaporative efficiency being of course 3·9 per centum better also. The economic evaporation of both, however, is immensely reduced, being 30 and 25 per centum less than when the rate of combustion was 12 pounds of anthracite per hour per square foot of grate surface.

With very low rates of combustion, say 5 pounds of anthracite per hour per square foot of grate surface, the vertical water-tube boiler gives an economic evaporation, and also a potential one, 8 per centum superior to those by the horizontal fire-tube boiler.

Under all rates of combustion, therefore, we perceive that the vertical water-tube boiler maintains a superiority both in the economic and in the potential evaporation over the horizontal fire-tube boiler; the shell, furnaces, and steam room being the same in each, and the best proportions given to both.

The horizontal fire tube boiler with a ratio of 25·288 square feet of heating surface to one square foot of grate surface, and 8·400 square feet of grate surface to one square foot of cross area of tubes for draught, the rates of combustion being 9·670 pounds of anthracite combustible per square foot of grate surface per hour, and 0·382 pound of anthracite combustible per square foot of heating surface per hour, evaporated under the atmospheric pressure, and from the temperature of 212° Fahr., 11·363 pounds of water per pound of anthracite combustible.

As the constituents of the "combustible" portion of the anthracite have a nearly invariable proportion among themselves, the "combustible" can be used in place of its carbon constituent without error relatively.

From the experiments on the boilers, in the accompanying table, made with anthracite, the following general conclusions can be drawn, namely:—

1st. *Of the influence of the rate of combustion on the economic evaporation.* It will be observed that whatever be the type of boiler, the slower the combustion, *ceteris paribus*, the higher the economic result from the combustible. And this conclusion holds good for rates varying from 4 to 20 pounds of combustible per square foot of grate surface per hour. It does not result from any more perfect combustion of the fuel, or any less air dilution of the products of combustion, but simply from the fact that the maximum heat absorbing surface given with the best types of boiler is too small to reduce to the same temperature in the same time a larger quantity of heated gases than that due to the above minimum weight of combustible per square foot of grate surface per hour.

2d. *Of the influence of the ratio of the heating to the grate surface.* With equal rates of combustion, the larger this ratio the higher in some proportion will be the economic evaporation by the fuel. With a combustion of 12 pounds of anthracite per square foot of grate surface per hour, there should not, for the best result, be less than 35 square feet of heating to one of grate surface with vertical water tube boilers; and not less than 45 square feet of heating to one of grate suaface with horizontal fire tube surface.

3d. *Of the influence of the calorimeter.* Considerable as is the influence exerted on the economic result by the rate of combustion, and by the ratio of the heating to the grate surface, more considerable still is the influence upon it exerted by the calorimeter, or proportion of draught area to the grate surface. The best calorimeter is $\frac{4}{31}$ of the grate surface; and it has this property, that it is the best for all types of boilers, for all rates of combustion, and for all ratios of heating to grate surface. The calorimeter may be reduced without loss, or much inconvenience to $\frac{1}{8}$th the grate surface, but it cannot be increased beyond $\frac{1}{7}$th without serious sacrifice of the economic result.

The horizontal fire tube boiler is much more sensibly affected by the calorimeter than the vertical water tube boiler, and requires a much nicer adjustment of it. It is also more affected by differences in the ratio of the heating to the grate surface, using the same rate of combustion.

There only remains to add, that the results of some late extensive experiments on the steamer "GEORGEANNA," made during her regular trips on Chesapeake Bay between Baltimore and Norfolk, with saturated steam, and with saturated and superheated steam combined according to WETHERED'S patent, have shown a very considerable gain given by the latter. These experiments have been conducted by a Board of Naval Engineers ordered by the Navy Department, and have been made with completeness and accuracy in every respect. The very considerable expense of making them has been borne by the proprietors of the line of steamers to which the "GEORGEANNA" belongs. Their public spirited President, Mr. FALLS, who has been untiring and successful in rendering these steamers the most economical of any in the country in the cost of power, having gratuitously placed them at the command of the department for experimental purposes. A repetition of the experiments on the "GEORGEANNA" is now in course on the "ADELAIDE," another steamer of the same line, and promises like results. These experiments have been made using the steam with three measures of expansion, namely, cutting off at 0·28, 0·45, and 0·65 per centum of the stroke of the piston from the commencement, and demonstrate with superheated steam, like similar experiments made with saturated steam, a scarcely perceptible difference in the economy of fuel due to these wide differences of expansion. They confirm in a remarkable degree, the facts and inferences relative to the use of steam very expansively, which will be found in this volume; and they will be given *in extenso*, with others made to determine the same physical truths, in the succeeding volume.

# EXPERIMENTS

ON

# SATURATED AND ADHEATED STEAM.

# COMPARATIVE EXPERIMENTS ON STEAM

# IN THE SATURATED STATE, AND ADHEATED

## ACCORDING TO WATERMANN'S SYSTEM:

### MADE WITH A VIEW TO DETERMINE THEIR RELATIVE ECONOMIC EFFICIENCY IN RAPPORT OF FUEL TO POWER DEVELOPED BY A STEAM-ENGINE.

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THE attention that, during the last few years, has been bestowed on the subject of superheating steam for use in steam-engines, and the great economic results announced to have been derived from it in England where the practice has reached its highest development, led MR. HENRY WATERMANN of New York City, a practical engineer and the manager of the Clinton Foundry, to devote much time to the study of the problem with the view of inventing a system of superheating which, while it should secure all the advantages, would avoid the disadvantages of the usual method.

The only method now in practical use consists in passing the steam, on its way from the boiler to the cylinder, through iron tubes or coils of iron pipe placed in the boiler uptake, where it is exposed to the temperature of the products of combustion as they immerge into the chimney. Among the disadvantages of this system is the greatly enlarged size necessary to be given to the uptake for the reception of the superheating apparatus, if the boiler calorimeter is to be retained, or the diminished draught consequent on the lessening of the calorimeter if no enlargement of uptake be made. Again, this method to be effective requires the products of combustion to be delivered upon the superheating apparatus at a temperature of at least 700° Fahr., which can only be done by largely reducing the economic evaporation of the boiler. The superheating apparatus is of itself objectionable from its cost and weight, and especially on account of the constant attention it requires from the engineer. It is always liable to accident by overheating when the engine is temporarily stopped, and it suffers a rapid deterioration of its tubes by internal corrosion. It must be guarded by a system of valves, and is obnoxious to all that can be advanced against the complication or addition of parts in machinery.

As the steam-pipe and cylinders are protected from external refrigeration by felting only, and as superheated steam, which acquires its superheated temperature with difficulty, loses it with ease and rapidity, a far higher temperature is required to be bestowed in the uptake than is retained in the cylinder. There

is, moreover, no practicable means of graduating the temperature, which will, of course, continually fluctuate with the furnace combustion and the quantity of steam used in a unit of time. It is found, too, that if the temperature of the steam is carried in the cylinder much above the degree normal to its pressure as saturated steam, a serious degradation of the metal will result, and the valve faces and seats become converted into graphite. The piston packing is also rapidly injured, and, if kept tight, has its friction much increased.

These disadvantages are probably sufficient, in a commercial view, to overbalance the slight gain obtained with the *average* of boilers, and will certainly prevent the method from obtaining permanency in ordinary practice. It may for a time be maintained, and with advantage, in some exceptional and very bad boilers —those which prime much and deliver their products of combustion at a very high temperature into the uptake—but the great and immediately appreciable benefits of simple mechanism will exclude it as soon as the interested efforts of patentees to introduce what is termed their "improvements" shall have ceased.

In the investigation of this subject WATERMANN repeated the experiments of FROST on superheated steam, using, like him, small inverted glass syphons with mercury for measuring pressure, and obtaining the superheating temperature from the boiling points of highly concentrated solutions of common salt in water. He was not aware at first of the cause that vitiated FROST'S experiments and produced his astonishing effects, and accordingly fell into a like error of opinion; but by later experiments with the same apparatus he detected this cause, which was the unsuspected presence of an insensible quantity of water retained in the liquid form on the surface of the tube by the strong attraction of the glass at temperatures but a little above 212° Fahr., and vaporized at the higher temperatures of the superheating. The whole quantity of water experimented with was but a minute drop—in fact a mere physical point—and the proportion of it which adhered to the surface of the tube in the liquid form at the lower temperature being large in comparison with the amount evaporated at that temperature, would, when evaporated at the higher superheating temperature, give an astonishing apparent increase of effect, the whole of which being credited to the mere fact of the superheating, a new discovery was erroneously thought to have been made in the physical properties of steam.

In WATERMANN'S experiments the effect given by the superheating ranged from two-and-a-half to three times that obtained from the saturated steam; and there appeared, also, this other anomalous circumstance, that the whole superheating effect was produced by a moderate amount of superheating, and could not be increased, except in a scarcely sensible degree, by a very great increase in the superheating temperature. The reason was that a moderate amount of superheating having vaporized *all* the water, the additional degree could only increase the bulk of the steam as a gas.

In the course of these experiments the practical sagacity of WATERMANN detected two facts which became the basis of his system. The first was the excessive sensibility to refrigeration of superheated steam, whereby a slight exposure deprived it of its superheat with amazing rapidity. Hence, he inferred, that in using it in the practical steam-engine it was indispensably necessary to protect the superheated steam from a less temperature than its own, the entire distance from the place of superheating to the end of the stroke of the piston. To give practical effect to this idea he proposed to steam-jacket the cylinder on both ends as well as on the sides, and to also steam-jacket the entire valve-chest and the steam-pipe between the superheating apparatus and the cylinder, taking care that the steam in the jackets had a somewhat higher temperature than the superheated steam they enveloped. The second fact he observed was, that a slight degree of superheating produced as high or very nearly as high a result as was derived from a much greater degree; and consequently that as the excessive temperature necessary with the uptake system was not a necessity

*per se*, all the advantages of superheating might be obtained with an economical boiler, could some mechanical means be devised whereby a slight superheating temperature could be given to the steam and maintained. He finally decided upon making the steam superheat itself by means of the differences of temperature due to differences of pressure produced by the use of a simple throttle-valve.

In this method the superheating apparatus is entirely outside of the boiler, and may be put in any convenient place, or within the steam-jacket of the cylinder if there be space enough. It consists essentially of a coil of pipe, or a faggot of tubes, arranged within a case and surrounded with the boiler steam. The steam-pipe connecting the apparatus with the cylinder valve-chest, together with the entire valve-chest and cylinder, are surrounded by steam-jackets filled also with boiler steam. At the point where the steam from the boiler enters the tubes or coils of the apparatus, an ordinary throttle-valve is placed, which, by being closed to the proper extent, will reduce the pressure and consequently the temperature to any desired degree. The steam to work the engine being drawn from within the tubes or coils of the apparatus will, evidently, not only enter the cylinder in a superheated state, but will continuously be exposed to additional superheating up to the moment of its discharge into the condenser. All the desirable practical conditions are thus successfully combined. The superheating given is but slight; and once obtained it will be retained, and perhaps increased, until the steam leaves the cylinder. The mechanical means are eminently simple and cheap; they require no attention, are attended by no danger, and allow the degree of superheating to be regulated by the same throttle which is used to control the speed of the engine. This system being independent of the temperature of the products of combustion, they can be reduced to the minimum, and a boiler of the highest economic evaporation employed in connexion with it.

The arrangement finally adopted by WATERMANN is the *ne plus ultra* of simplicity. It consists in making the entire steam-pipe the superheating apparatus, by enclosing it in an outer pipe and filling the annular space between with boiler steam. The throttle-valve is, of course, placed at the boiler end of the steam-pipe, and the cylinder and its valve-chest are jacketed and the jackets filled with boiler steam also. If it be determined to use superheated steam at all, this is certainly the best method in every respect, and is so elementary and efficient as to scarcely leave room for further improvement.

To practically test the value of these ideas, the small steam-engine on which the experiments hereinafter detailed were made was constructed by WATERMANN at his foundry. In this he was assisted by the liberality of MR. GEORGE V. HECKER, the eminent flour manufacturer of New York City, an amateur engineer of considerable experience, who provided the funds and gave the benefit of his advice. Before commencing the experiments an application was made by these gentlemen to the Navy Department for a Naval Engineer to superintend and report upon them, and the writer was selected for this service by the Hon. ISAAC TOUCEY, who at that time was the Secretary of the Navy.

In the following pages will we found a description of the machinery and of the manner of making the experiments, followed by the experimental data and a discussion of the results.

In order to distinguish WATERMANN'S system from the uptake and other systems of superheating, the term ADHEATED has been used instead of SUPERHEATED; and with a view, also, to indicate the far less degree of temperature employed, its theory being to communicate only so much additional heat to the saturated steam as will preserve it, with the aid of the steam-jackets, in the gaseous form until it is discharged into the condenser; all idea of deriving any gain from increasing the volume of steam as a gas by superheating it being abandoned.

The experiments were made at the Clinton Foundry, No. 239 Cherry Street, New York City; and during their progress were witnessed by many practical engineers and gentlemen of scientific pursuits.

## ENGINE.—(Plate I.)

The experimental engine consisted of a single direct-acting cylinder, using the steam with condensation and expansion: its power was solely employed in revolving a fan in the open air of a closed apartment.

The cylinder was placed vertically beneath the engine shaft, and fitted at the centre of its length with a common three-ported short slide steam-valve, unpacked, on the flat back of which there was placed a cut-off slide not variable. The ports of the cut-off valve were through the steam-valve, and were closed when the piston had completed 19 per centum of its stroke from the commencement, so as to allow the steam to act expansively through the remaining 81 per centum. The steam-valve had lap on the steam side which cut off at 83 per centum of the stroke of the piston from the commencement, leaving the steam to act expansively through the remaining 17 per centum. Both steam and cut-off valves were worked direct by eccentrics on the engine shaft.

The piston was packed with a stiff thick brass ring, cut obliquely through and sprung into the cylinder.

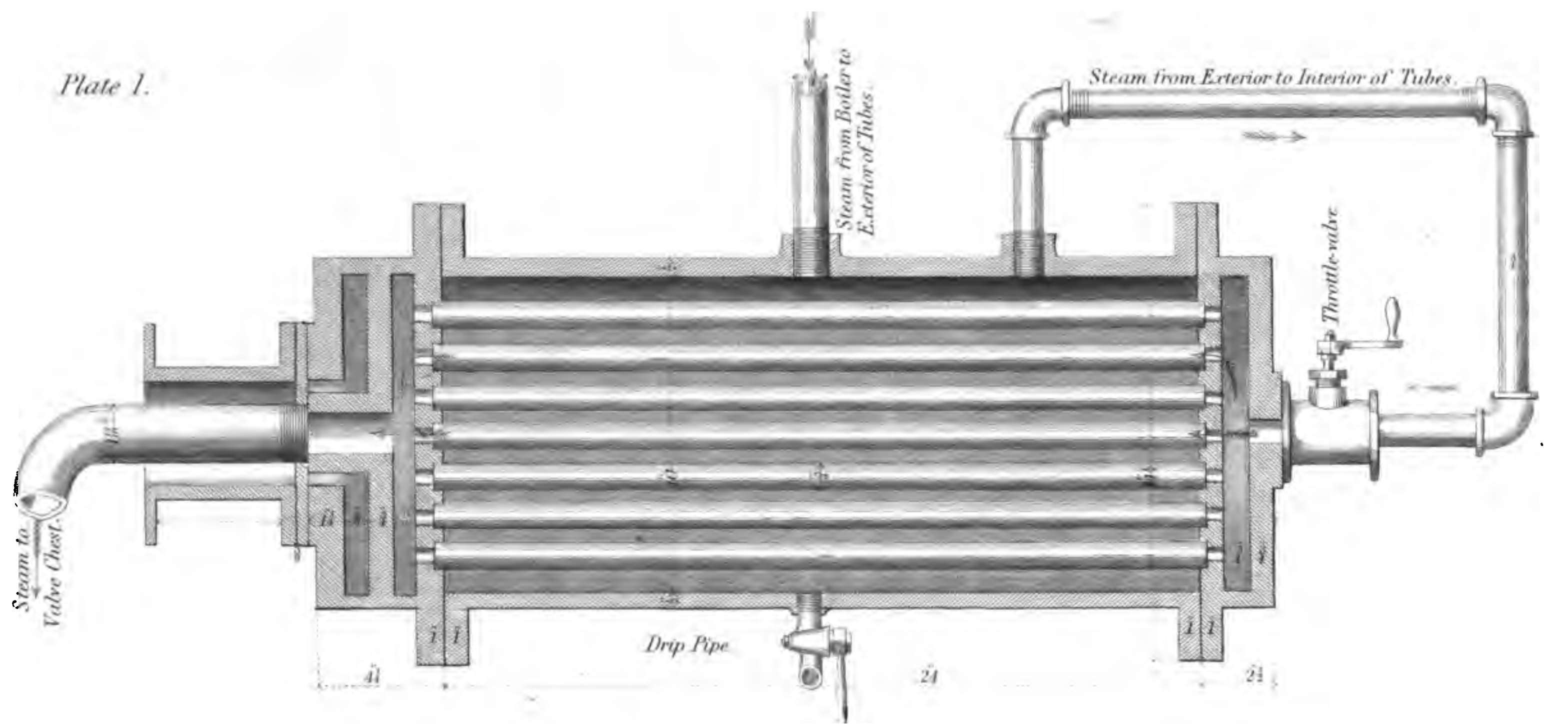
The cylinder-cover had a neck 9½ inches long between clyinder and piston-rod stuffing-box; the piston-rod worked through the neck, which was required in order to carry the rod through the cylinder-jacket. Within a similar envelope or neck the two valve-stems were carried through the cylinder-jacket. The piston-rod was attached to a crosshead moving between guides; from the crosshead a connecting-rod 22 inches long between centres extended directly upwards to the crank-pin. Upon the engine shaft there were placed a fly-wheel, the two eccentrics for the valves, the pulley for the governor, and the driving-drum from which, by means of a belt, the working-fan was driven. The air and feed pumps were worked from a pin in the periphery of the hub of the fly-wheel. During the experiments the diameter of the driving-drum was varied, but the diameter of the fan-pulley remained constant.

The air-pump was vertical, open-topped, and single acting; it was inserted within the condenser which was of the common jet kind, and delivered into a hot-well immediately over it. The pump had no foot-valve; the disc-valve in its piston and the delivery-valve at the hot-well, both guided on the piston-rod, were all the valves. The injection-water was derived from the Croton Aqueduct, and came in under a very great head through a pipe controlled by a cock.

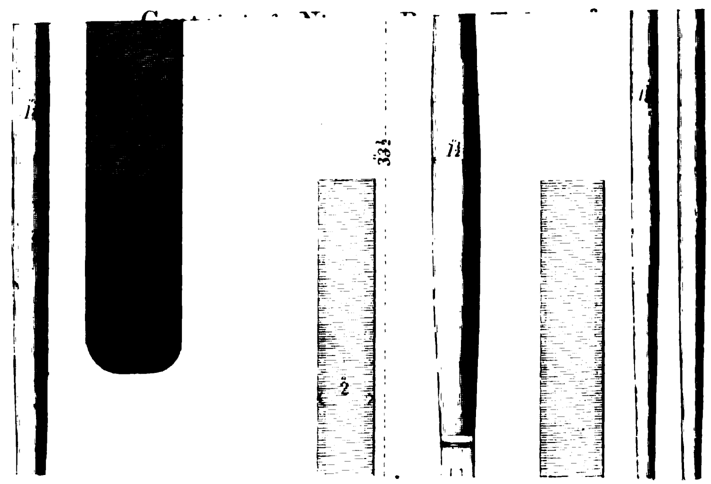
From the hot-well the injection-water and water of condensation of the steam flowed through a pipe three feet long to the receiving tanks beneath, which were two open-topped, water-tight, cubical-shaped vessels, placed side by side, with an overflow or weir from each into the other. Upon the bottom of each tank was a metal water-tight valve controlling the orifice of the discharge pipe. The pipe from the hot-well was so arranged that it delivered at will into either tank, so that while the one was being emptied the other was filling. The time required for emptying was a very small fraction of that required for filling, and as a tank was emptied. its bottom valve was closed, and the other tank allowed to fill till it discharged over the weir. Each tank was a parallolepipedon with a small hopper head at the weir. The following were the dimensions, namely:—

| | |
|---|---|
| Length, . . . . . . . . . | 46¼ inches. |
| Breadth, . . . . . . . . | 18¾ " |
| Depth, . . . . . . . . . | 20¾ " |
| Capacity, including the hopper-head of 7½ by 7½ by 3½ inches, . | 10·180 cubic feet. |

Plate 1.




TUBULAR ADHEATER,




Belt.

The feed-pump was a vertical plunger-pump, and drew its water from an open-topped, water-tight, wooden tank of the following dimensions, namely:—

| | |
|---|---|
| Length, | 15 inches. |
| Breadth, | 14¼ " |
| Depth, | 20¼ " |
| Capacity, | 2·580 cubic feet. |

This tank was not supplied from the hotwell, but from the Croton Aqueduct, and had, of course, its temperature. The feed-pump communicated with its bottom by means of a pipe and stop-cock. All the water pumped into the boiler was first accurately measured in this tank as follows, namely:—It was exactly filled through a pipe with a stop-cock and very rapidly, the feed-pump being shut off during the operation. The cock in the supply-pipe was then shut, and that in the feed-pipe opened. When all the water had been pumped out, the feed-pipe stop-cock was closed, and that in the supply-pipe opened, and the operation repeated.

The entire steam-cylinder, including top, bottom, sides, valve-chest, and the necks enveloping the piston-rod and valve-stems, was placed within another cylinder of 19¾ inches inner diameter and 21¼ inches outer diameter, with an interior height of 28⅝ inches and an exterior height of 30⅝ inches, exposing an exterior surface of 19·1217 square feet. This outer cylinder formed a casing or jacket around the steam-cylinder and its appurtenances, leaving a clear space on every side of them that could be filled with either steam or air at will. In the top of this outer cylinder were the stuffing-boxes for the piston-rod and valve-stems, &c. It had a hole of two inches diameter fitted with a screw-plug, by the removal of which the interior could be exposed to the atmosphere. This hole was the sole communication with the air; consequently when open there could have been but a very slight circulation within. A branch, with a stop-cock, of the main steam-pipe was screwed into the top of this outer cylinder which, through it, could be filled at will with steam of very nearly the boiler pressure. In the bottom of the outer cylinder was screwed a small drip-pipe with stop-cock, for draining off the water of condensation.

Three instruments were used for experimenting with adheated steam, namely, the coil, the tubular adheater, and the steam-jacketed steam-pipe.

And first, of the Coil. Within the space between the steam-cylinder and its jacket, and around the cylinder and valve-chest, there was coiled what may be considered an extension of the steam-pipe. It consisted of a wrought iron pipe 1¼⅛ inches in exterior diameter, 1½ inch in interior diameter, and 24 feet in length. This coil contained 9·4224 square feet of surface measured on the inside, and 11·3861 square feet measured on the outside: its capacity was 0·2945 cubic foot. One end, situated near the top of the jacket, was open and fitted with a butterfly throttle-valve controlled by the engine governor; the other end was secured to the bottom of the valve-chest into which it delivered. When the coil was used, the cylinder jacket was filled with steam of very nearly boiler pressure, which, entering the upper end of the coil, was reduced by the throttle-valve to any desired degree, and, after traversing the coil at this reduced pressure, was delivered into the valve-chest. By this arrangement the steam within the coil had a less temperature than that without, and underwent an adheating in proportion to the difference of temperature, extent and quality of surface, and time of exposure. At the same time, the entire exteriors of the steam cylinder, of the valve-chest, and of the enveloping necks for the piston-rod and valve-stems, were also adheating surface. The greater thickness of the metal, however, with these surfaces, averaging about ⅞-inch, would make them much less efficient adheaters than the coil surface.

Second, of the Tubular Adheater. This was a separate vessel placed outside of the cylinder-jacket. It

was a cast iron horizontal cylinder 24 inches long, 11 inches outside diameter, and 10 inches inside diameter. It had two cast iron hollow heads bolted to its flanges. The inner sides of the heads were drilled, and composed the tube plates. The tubes, filling the interior of the adheater, and constituting its adheating surface, were of brass, ninety in number, $\frac{3}{4}$-inch in external diameter, $\frac{11}{16}$-inch internal diameter, and 24 inches length between plates, total length $24\frac{3}{4}$ inches. Area of adheating surface measured on the outside of the tubes 35·3400 square feet. Capacity of tubes 0·4640 cubic foot. Area of external surface of adheater for radiation 7·0782 square feet. The steam from the boiler was delivered into the top of the adheater, and the space between the cylindrical shell, tube-plates, and exterior of the tubes was kept filled with steam of sensibly the boiler pressure, and this space constituted the exterior compartment of the adheater. From the top of the adheater the steam was carried by an elbow-pipe to the centre of one of the hollow heads, which communicated with the interior of the tubes, and in this pipe and near the head the throttle-valve was placed, by means of which the pressure of the steam was reduced to any desired degree within the tubes. The tubes delivered the steam of reduced tension into the other hollow head, whence it passed by a short steam-pipe directly to the upper part of the valve-chest through the cylinder-jacket. This short pipe was jacketed from the adheater to the cylinder-jacket. The hollow heads and the interior of the tubes constituted the interior compartment of the adheater. By this arrangement the exterior compartment being filled with steam of sensibly the boiler tension, and the interior compartment being filled with steam of less pressure to any desired extent, the latter would become adheated in proportion to the difference of temperature of the two pressures, extent and quality of surface, and time of exposure. When using the adheater, the cylinder-jacket could be filled with either steam of the boiler pressure or air at will. When the jacket was filled with steam, the short pipe connecting it with the adheater, and the entire steam-cylinder and appendages, were additional adheating surfaces. When the adheater was used the coil was removed, and the opening by which it delivered to the bottom of the valve-chest plugged up. At the bottom of the adheater there was a small drip-pipe and stop-cock to drain off the water of condensation from the exterior compartment.

Third, of the Steam-jacketed Steam-pipe. The steam-pipe was converted into an adheating instrument by simply enveloping it in another pipe of greater diameter, and filling the annular space between them with steam of the boiler pressure. The outer pipe and the cylinder-jacket were in free communication. The throttle-valve was placed within the steam-pipe at its boiler end, and between the boiler and steam-pipe a priming box was interposed to intercept any water that might pass over from the boiler in the solid state. By a proper closure of the throttle-valve any desired reduction of steam-pressure and temperature could be made in the steam-pipe from that in the annular space, and the interior steam would be thus adheated to a degree corresponding with the difference of temperature, extent and quality of surface, and time of exposure.

The length of the steam-jacketed steam-pipe was 15 feet; its inner diameter 1 inch; outer diameter $1\frac{1}{4}$ inch; outside surface 4·9090 square feet; capacity 0·081802 cubic foot; outside diameter of casing-pipe $2\frac{3}{4}$ inches.

The cylinder-jacket, with the exception of the top; the steam-jacket of the steam-pipe; and all the connecting parts and branches of pipes, cocks, &c., were well covered with thick felt. The tubular adheater was immersed in a box of sawdust.

When the adheating instruments were used, all the water of condensation in the steam-jackets and tubular adheater were drawn off by cocks, and with it all the water that might have been primed over from the boiler, or carried over in the vesicular state, so that nothing but pure dry steam could enter the cylinder,

the cylinder-jacket and the exterior compartment of the tubular adheater acting as traps or priming boxes for the separation of the water from the steam.

The engine was also arranged for using the cylinder-jacket filled with steam, but not in combination with any adheating instrument, by bringing a branch from the main steam-pipe through the cylinder-jacket and delivering into the valve-chest, the throttle-valve being situated in the branch just outside the jacket. Under these conditions all the water of condensation from the steam-pipe, and all the water brought over mechanically from the boiler, entered the cylinder with the steam. This arrangement was varied by leaving only a short piece of the steam-pipe attached to the valve-chest, and placing the throttle within its open end, the whole being inside the cylinder-jacket as the coil was. This permitted the cylinder to be supplied with pure steam, the jacket acting as a trap for the separation of the water.

From the preceding description it will be seen that the following set of experiments could be made, viz:—

1st. On saturated or common steam used with air in the cylinder-jacket.

2d. On steam adheated in the tubular adheater and used with air in the cylinder-jacket.

3d. On saturated steam used with steam in the cylinder-jacket; and under the two conditions of delivering the steam directly from the boiler into the valve-chest, or indirectly by first delivering it into the cylinder-jacket whence it entered the valve-chest through the short pipe.

4th. On steam adheated in the coil and used with steam in the cylinder-jacket.

5th. On steam adheated in the tubular adheater and used with steam in the cylinder-jacket.

6th. On steam adheated in the steam-jacketed steam-pipe and used with steam in the cylinder-jacket.

In each case the experiments admitted of variation by the employment of different pressures, different degrees of throttling, different speeds of piston, and of the two measures of expansion permitted by the valve gear.

The power of the engine was employed in rotating a common fan freely in the open air of a closed room. This fan had the form of a common paddle-wheel, and was composed of five blades or paddles, each 12 inches long by 12¾ inches wide. Outside diameter of blades 4 feet. On the fan-shaft was a cast iron pulley of 9⅝ inches diameter, which remained throughout all the experiments, and was driven by a belt from the driving-drum on the engine shaft. This belt was provided with a tightening pulley.

The following are the principal dimensions of the engine:—

| | |
|---|---|
| Diameter of cylinder, | 5¼ inches. |
| Stroke of piston, | 10 " |
| Diameter of piston-rod, | 1⅛ inch, |
| Space displacement of piston per stroke, exclusive of rod, | 0·1224 cubic foot. |
| Area of cylinder port (⅝ by 3 inches =), | 1·875 square inch. |
| Length of steam passage from valve face to cylinder (for one end of cylinder), | 6·25 inches. |
| Clearance, | $\frac{5}{16}$-inch. |
| Area of port through steam-valve for cut-off (⅜ by 3 inches =), | 1·125 square inch. |
| Length of passage through steam-valve, | 1·375 inch. |
| Steam space between piston at end of stroke and face of steam-valve (at one end of cylinder), | 17·845 cubic inches. |
| Steam space in the cut-off passage through the steam-valve, | 1·547 cubic inch. |
| Bulk of steam exhausted from the cylinder per stroke of piston, | 0·13272 cubic foot. |
| Diameter of air-pump, | 4 inches. |
| Stroke of air-pump piston, | 4 " |
| Diameter of feed-pump, | ⅞-inch. |
| Stroke of feed-pump piston, | 3 inches. |

## BOILER.—(Plate II.)

The boiler is of the common locomotive type, with rectangular furnace and cylindrical barrel of 18 inches diameter, and with tubes extending horizontally directly from the furnace to the uptake. The uptake is cylindrical like the barrel, of which it is, in fact, an extension, and from its top a sheet iron pipe of 12 inches diameter conducts the products of combustion to the chimney of the establishment. A large door in the end of the uptake allows the sweeping of the tubes, removal of ashes, &c. The ash-pit, like the uptake, is a separate construction of sheet iron, and is likewise merely placed in connexion with the boiler shell.

The furnace grate was constructed 19 by 20 inches, giving an area of 2·64 square feet, but proving much too large for the quantity of steam required, its width was reduced, by the insertion of bricks on each side, to 10 inches, the length remaining 19 inches as before: the area was thus made 1·32 square foot, and remained so during most of the experiments. In some of them it was enlarged by taking out the bricks on one side, which increased its area to 1·98 square foot. The mean area for all the experiments was 1·4707 square foot. The top of the furnace was flat; the water ways at front and back were 2½ inches in width from outside to outside of metal, and at the sides of the furnace 2 inches. The furnace door was 11 inches high by 10 inches wide, and it was not perforated with air apertures. The height of the furnace was 17½ inches above the top of the grate, and its capacity above the same level was 3·30 cubic feet.

The tubes were of iron; they were twenty-two in number, each was 3 feet in extreme length, 1½ inch in outside diameter, and 1·356 inch in inside diameter, weight 3½ pounds. They were distributed in four horizontal rows, the tubes of each row being placed above the spaces between the tubes of the row immediately beneath. Between the tubes there was in all directions a clear water space of 1 inch, making the axes of the tubes 2½ inches apart. The axes of the top row of tubes were in the horizontal diameter of the barrel.

The steam-drum of 12 inches diameter was placed immediately above the centre of the furnace, and from its top, at a height of 16 inches above the top of the boiler, the steam-pipe to the cylinder was inserted.

The water line was carried at 5 inches above the top of the furnace.

The grate-bars were ⅝-inch wide with air spaces ⅜-inch wide between them. Aggregate area for admission of air 71 square inches.

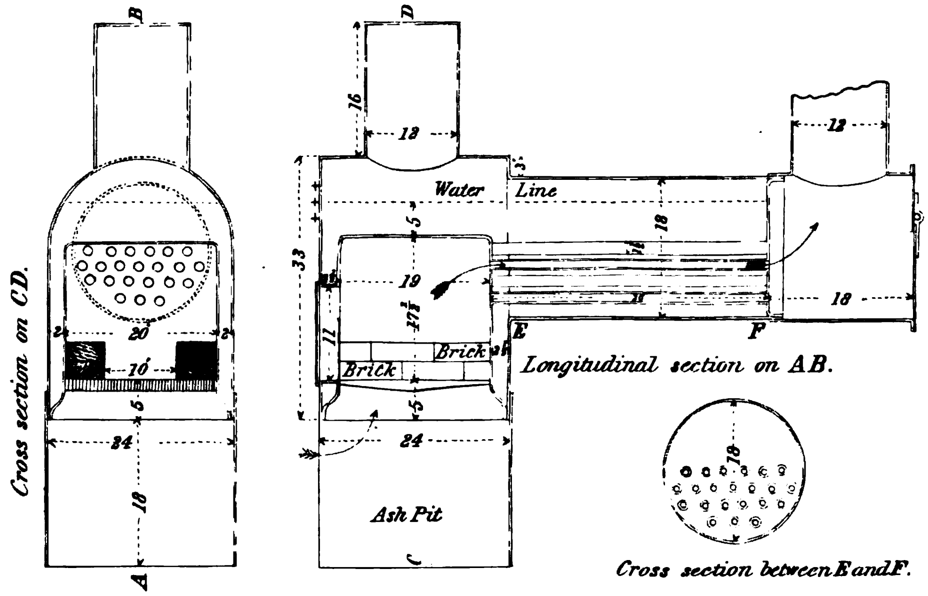
The steam-pipe connecting boiler and cylinder was 15 feet long, 1¼ inch outside and 1 inch inside diameter; capacity 0·0818 cubic foot.

The steam-pipe and boiler were thoroughly covered with thick new felt, coated with an anti-conducting preparation.

The following are the principal dimensions and proportions of the boiler required to be known, namely:—

| | |
|---|---|
| Area of fire-grate. (Mean for all experiments,) . . . . | 1·4707 square foot. |
| " heating surface in furnace, . . . . . | 10·02 square feet. |
| " " tubes (calculated for the interior circumference), | 23·43 " |
| Total area of heating surface, . . . . . . | 33·45 " |

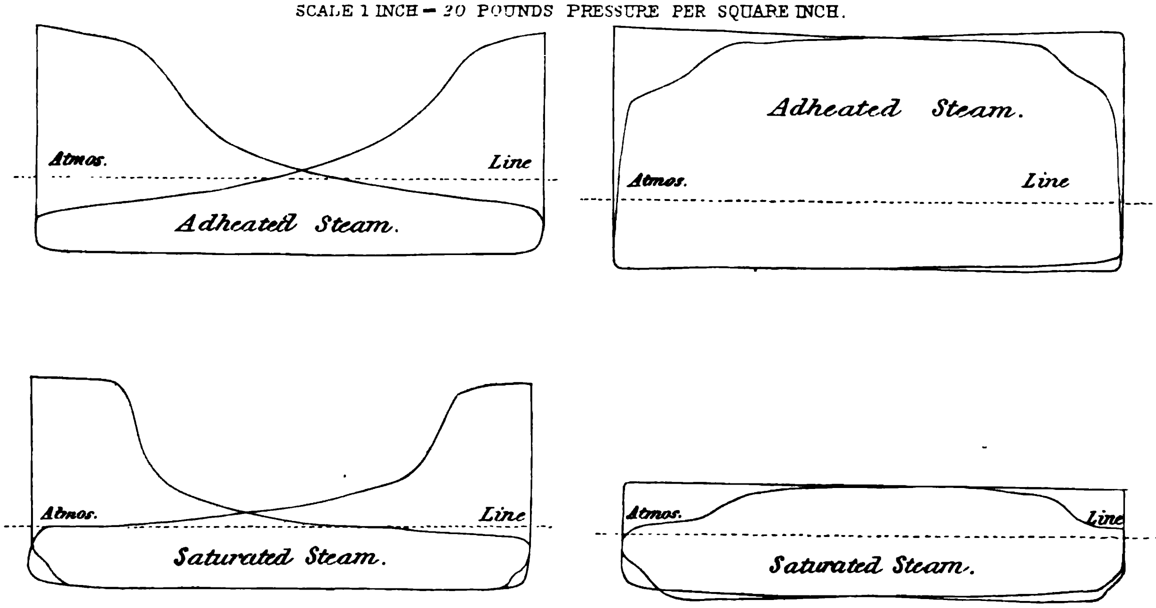
# BOILER FOR WATERMAN'S ENGINE.




## INDICATOR DIAGRAMS

*From WATERMAN'S EXPERIMENTAL ENGINE, Showing the Distribution of the Steam in the Cylinder during the Experiments with Adheated and Saturated Steam.*

SCALE 1 INCH = 20 POUNDS PRESSURE PER SQUARE INCH.



| | |
|---|---|
| Aggregate cross area of tubes (for draught), | 0·2206 square foot. |
| Capacity of steam room, | 2·464 cubic feet. |
| Weight of water at 212° Fahr., | 425·000 pounds. |
| Capacity of steam room in boiler and steam-pipe, | 2·546 cubic feet. |
| Ratio of heating to grate surface, | 22·744 to 1·00. |
| " grate surface to cross area of tubes, | 6·667 to 1·00. |

With the machinery as described the experiments were conducted in the following manner:—

## MANNER OF CONDUCTING THE EXPERIMENTS.

In making the experiments, steam was first raised to the intended working pressure by wood, and the engine operated for several hours to bring it into uniform action. The wood was then allowed to burn down to the few coals required for kindling the free burning semi-anthracite used, care being taken to maintain the boiler pressure by throttling. The weighing of the coal was now commenced, and the quantity given in the following Table, No. 1, is the whole weight consumed during each experiment. The coal was allowed to burn down at the end of the experiment as much as possible, and still maintain the speed of the engine to the last. As soon as the time expired, the coal that remained in the furnace was withdrawn and wet out, and the unburnt lumps were carefully picked out and weighed. The ashes, &c., withdrawn from the ash-pit, were accurately weighed dry. The same weighing apparatus—one of Fairbanks' small platform scales—was employed throughout.

The smoke-pipe was provided with a damper worked by the boiler pressure, which was thus maintained with great uniformity, the widest variations not exceeding a few pounds. The damper was generally much closed, as the supply of steam required in the greater number of the experiments fell considerably below the capacity of the boiler to furnish.

In all the experiments previous to June 29th, one-half the grate surface was bricked off by a row of bricks on each side, extending from the front to the back of the furnace; but during the experiments after June 29th one of these rows was removed, and the grate surface previously in use was thus increased fifty per centum.

Throughout all the experiments the water was maintained at the same height in the boiler by a fixed mark on the glass gauge; and each experiment began and ended with the water *precisely* at this level. In addition to the glass gauge, there were three cocks for verifying the water level, and assuring that the communications to the glass gauge were not obstructed.

The boiler was supplied with water by the engine feed-pump from a tank, in which it was previously measured. This tank was filled from the Croton Aqueduct; the filling occupied only three minutes, which was so very small a proportion of the time required for emptying it, that the effect upon the water level in the boiler was insensible. Of course, the feed-pump was shut off during the time of filling. The feed-pipe was inserted in the bottom of the tank, and drained it completely dry. A thermometer kept constantly immersed in the tank, gave at all times the temperature of the feed-water.

The injection-water (from the Croton Aqueduct) and the water formed by the condensation of the steam, drawn from the condenser by the air-pump, was discharged into two tanks, alternately, one being in progress of filling while the other was being emptied. These tanks were of precisely the same dimensions, and were situated side by side. They were emptied from the bottom completely dry. A short pipe connected the hot-well of the pump with the tanks, and a thermometer, kept constantly immersed in the stream flowing from this pipe, gave at all times the temperature of the water thrown out by the air-pump.

The water formed by the condensation of the steam in the cylinder-jacket, steam-pipe, and tubular adheater was constantly drawn off through one pipe and cock for the jacket and steam-pipe, and through another pipe and cock for the adheater. The pipes being placed at the lowest points, and the cocks being kept slightly open, the pressure from within discharged the water as fast as it formed and without discharging steam with it. A little practice enabled these cocks to be so graduated that neither water remained nor steam came out. The water thus obtained was accurately weighed on the scales. Care was taken at the commencement and end of each experiment that the jacket and adheater should be completely free of water.

An excellent indicator remained permanently attached to the cylinder, and was put in communication with both ends by a two-way cock with pipes of large diameter. This instrument was absolutely correct, and the pencil would trace the same line as long as the steam remained unchanged.

A barometer was kept suspended in the engine room.

The pressure of the steam in the boiler, in the cylinder-jacket, and in the cylinder valve-chest, was determined by manometer gauges manufactured by the *Novelty Iron Works*. They were large fine instruments, and, during the experiments, were frequently compared with a standard for correction. The steam pressure in the cylinder was controlled by the engine governor, and the speed of the piston was thus made very uniform.

In addition to the thermometers immersed in the feed and injection-water tanks, one was kept suspended half way between the engine and boiler, but shielded from the radiation of either. Its indications were registered for the temperature of the engine room. The feed-water tank was situated opposite and between the engine and boiler also. A thermometer was inserted deeply into the top of the cylinder-jacket; a large portion of the stem as well as the bulb was immersed, and its indications were registered for the temperature of the steam or air within. A thermometer was inserted in the tubular adheater in the centre of the current of adheated steam leaving it on the way to the cylinder; it was placed in such a position as to be shielded from the effects of radiation from the boiler steam surrounding the adheated steam. The temperature of the injection-water entering the condenser was obtained from a pipe outside the engine room, and exposed to the temperature of the external air.

The number of double strokes made by the engine-piston was taken by a counter worked by the engine.

The relative speed of the engine and fan was regulated by pulleys and a connecting belt. The pulley on the fan shaft remained throughout of the same diameter, and only the diameter of that on the engine shaft was varied. The belt was provided with a tightening weight, and frequent examination failed to detect the least slip.

The vacuum in the condenser was taken by an open mercurial syphon gauge.

Each experiment, once commenced, was continued without interruption until completed. Most of the experiments were for either sixty or thirty consecutive hours; one was for seventeen hours and two were for fifteen hours each. During each experiment observations were recorded at the end of every hour of the number on the face of the counter, of the steam pressure by the three manometer gauges, of the temperature by all the thermometers, of the vacuum gauge, and of the barometer. An indicator diagram was taken at the close of each hour from both ends of the cylinder, so that there were as many double diagrams taken as there were hours during the experiment. Not only was the number of feed and injection-water tanks filled noted, but the exact time of filling each was recorded in the table of observations, so as to serve as a check on error. For the same reason, the number of double strokes made by the engine-piston per minute was worked out and entered opposite each hour. The coal was weighed out in nearly

equal quantities, and the time of consuming each quantity was noted as a check both upon the weight of fuel and upon the regularity of the firing. The water of condensation drawn off from the cylinder-jacket and from the tubular adheater, was weighed at hourly or two hourly intervals for the purpose of checking against error, and of detecting any irregularity in the quantity.

The coal used was good Trevorton semi-anthracite of small egg size, screened, and of very uniform quality. It was fired with great regularity.

It was purposed to make these experiments with almost laboratory accuracy, and no pains were spared to secure it. It was also aimed to conduct them with complete uniformity in all respects from beginning to end, so that whatever there might be of disturbing cause should act with regularity, and affect as little as possible the comparative correctness of the results. The observations and the indicator diagrams were very uniform throughout each experiment, the extremes differing but slightly; and it is confidently believed that the means of so many, during experiments of such length, taken with such instruments and with so much care, may be depended on for an extremely close approximation to the truth; far closer, indeed, than can be of any practical importance.

The observed and calculated data of the experiments are given in the two following Tables, No. 1 and No. 2, which are succeeded by a discussion and generalization of the results. Each table is preceded by an explanation of its contents.

### EXPLANATION OF TABLE No. 1, CONTAINING THE DATA AND RESULTS OF THE EXPERIMENTS MADE WITH THE EXPERIMENTAL ENGINE OF HENRY WATERMANN, AT 239 CHERRY STREET, NEW YORK CITY, TO DETERMINE THE COMPARATIVE ECONOMY IN RAPPORT OF FUEL AND POWER OF SATURATED AND ADHEATED STEAM UNDER VARIOUS CONDITIONS OF PRESSURE, EXPANSION, AND ADHEATING.

In the columns of the following table will be found the Observed Data and the Calculated Results of all the experiments made with the precedingly described apparatuses. For the convenience of the reader, the headings of the columns have been made so fully descriptive that he may not, in order to understand their contents, be interrupted by the necessity of referring to other parts of this paper. It will be observed that the experiments are not arranged in the order of their dates, but according, so far as their nature permitted, to their natural connexion. For brevity of reference the columns are designated by letters, and the lines by numbers.

SATURATED STEAM USED WITH AIR IN THE JACKET. The experiments recorded in columns from A to H, both inclusive, were made with saturated or common steam, the cylinder-jacket being filled with atmospheric air. The top of this jacket was perforated with a two inches diameter hole, through which the air had both ingress and egress, so that its circulation must have been extremely slow. The steam-pipe and jacket, with the exception of the top of the latter, were thoroughly felted. During the experiments in columns B, D, and F, the cylinder and valve-chest within the jacket were also covered with felt two inches thick. During the experiments in columns A, C, E, G, and H, the cylinder and valve-chest within the jacket were without felting; and this was their condition throughout all the remaining experiments. The steam was taken direct from the boiler to the valve-chest, so that all the water due to condensation in the pipe, or passed in the solid state from the boiler, entered the cylinder with it. The throttle-valve was placed within the steam-pipe just outside of the cylinder-jacket, and was controlled by the governor of the engine.

During all the experiments the fan shaft pulley, to which the power was applied, remained of the same

diameter, namely, 9⅝ inches; but the engine shaft pulley, from which the power to drive the fan was taken, was varied in diameter, being 24 inches in columns D and H, 30 inches in columns A, B, and E, and 36 inches in columns C, F, and G. In consequence of this difference in the diameters of the engine shaft pulley—the diameter of the fan shaft pulley remaining constant—there was a difference in the velocity of the piston and in the average steam pressure upon it, when about equal weights of steam were drawn from the boiler in equal times.

The conditions of the experiments were still further varied by using the steam with different measures of expansion. During the experiments in columns A, B, C, and D, the steam was cut off in the cylinder at 0·19, while during the experiments in columns E, F, G, and H, it was cut off at 0·83 of the stroke of the piston from the commencement.

Adheated Steam used with Air in the Jacket. The experiments recorded in columns I and J were made with the cylinder-jacket filled with atmospheric air, as described in the immediately preceding section; but the steam used in the cylinder, instead of being saturated, was previously adheated in the tubular adheater many degrees above the temperature due to its pressure.

The steam from the boiler was delivered into the outer compartment of the adheater, whence it was carried through an elbow-pipe containing the throttle-valve into the interior compartment, from which it passed through a short steam-pipe to the valve-chest. This short steam-pipe was encased in an outer one, and the annular space between them being open to the cylinder steam-jacket, was filled with its steam. By this arrangement all the water resulting from the condensation of steam in the pipe leading from the boiler to the adheater, and in the adheater itself, as well as any water that might pass over from the boiler in the solid state, lodged in the outer compartment, which thus acted as a priming-box, and was continuously drawn off by a cock beneath. Now, it is evident that if steam of the boiler pressure be in the outer compartment of the adheater while, *by the action of the throttle-valve*, steam of many pounds less pressure be in the interior compartment, the steam of less pressure, thus surrounded by the steam of greater pressure, will have its temperature considerably increased above that which is normally due. In these experiments, then, the cylinder, having its jacket filled with atmospheric air, was supplied with dry steam adheated on entering the valve-chest many degrees above the temperature due to its pressure as saturated steam.

The adheater and its connecting pipes were well felted; the engine shaft pulley was 30 inches in diameter,—the fan shaft pulley being, as before, 9⅝ inches in diameter; and the steam was cut off in the cylinder at 0·19 of the stroke of the piston from the commencement. The only difference in the conditions of the two experiments was, that in one the boiler pressure and the back pressure against the piston were higher than in the other.

Saturated Steam used with Steam in the Jacket. The experiments recorded in columns K to N, both inclusive, were made with saturated steam admitted to the cylinder, but with the latter steam-jacketed; that is to say, the cylinder, its valve-chest, and the piston-rod stuffing-box, were surrounded by steam of nearly the boiler pressure. The steam was cut off in the cylinder at 0·19 of the stroke of its piston from the commencement. The diameter of the pulley on the engine shaft was 30 inches,—of that on the fan shaft 9⅝ inches; consequently the fan made ($\frac{30}{9⅝}$=) 3·1169 revolutions for each double stroke of engine piston. The steam-pipe and the cylinder-jacket, with the exception of its top, were thoroughly felted.

The conditions of the experiments recorded in columns K and L differed from those recorded in columns M and N in the manner in which the steam was obtained from the boiler. In experiments K and L the

JRE. EXPANSION. AND ADHEATING.

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steam from the boiler was delivered first into the jacket, and thence passed through a short pipe containing the throttle-valve into the valve-chest. This pipe and its throttle were within the jacket, and the valve was controlled by the governor of the engine through a stuffing-box. By this arrangement all the water resulting from the condensation of steam in the steam-pipe leading from the boiler to the jacket, or passed over in the liquid state from the former, was deposited in the jacket as a priming-box, and continuously drawn off with the water of condensation due to the jacket itself. Hence, no water could enter the cylinder which was thus supplied with pure saturated steam alone.

In experiments M and N the steam from the boiler was delivered direct into the cylinder valve-chest, carrying with it all the water resulting from the condensation of steam in the steam-pipe, or passed over from the boiler in the liquid state. Hence, the cylinder received, in addition to the saturated steam, some liquid water. The cylinder-jacket was filled with steam of nearly boiler pressure through a small, short, independent pipe which was a branch of the main steam-pipe. The throttle-valve was placed in the main steam-pipe, just outside the jacket, and was controlled by the engine governor. It will be observed in all the experiments that, as the steam pressure in the cylinder was much less than the boiler pressure, the surfaces of the cylinder and valve-chest acted as an adheater to the steam within.

Adheated Steam used with Steam in the Jacket. All the experiments recorded in columns from O to Z, both inclusive, were made with "Adheated Steam used with Steam in the Jacket," but they varied greatly in the conditions of pressure, diameter of engine shaft pulley, condensation, measure of expansion employed, and degree and manner of adheating. The steam was adheated in three different ways, namely, in the coil, in the tubular adheater, and in the steam-jacketed steam-pipe. The following is a description of each method :—

Adheated in the Coil. In the experiments embraced in columns O to R, both inclusive, the steam was adheated in the coil of iron pipe surrounding the cylinder and contained in the steam-jacket. This pipe was of the same diameter as the steam-pipe, of which it was, in fact, an extension. The steam was delivered from the boiler first into the cylinder-jacket as a priming-box, and was thence passed through the coil into the cylinder valve-chest. The throttle-valve—controlled by the engine governor—was placed in the receiving end of the coil. By this arrangement the following effects were produced, namely :—

1st. The jacket being constantly filled with steam of nearly the boiler pressure, and the coil being constantly filled with steam of less pressure and consequently of less temperature,—made so to any required degree by the mere action of throttling,—the steam of less temperature within the coil was exposed to the steam of greater temperature within the jacket, and there inevitably followed the degree of adheating due to the difference of temperature, area of surface in the coil, time of exposure, &c. By this simple method the temperature of the steam within the coil due to its pressure as saturated steam, could be considerably increased.

2d. All the water resulting from condensation in the steam-pipe, or carried over from the boiler in the liquid state, was deposited in the jacket and drawn off continuously with the water of condensation due to the jacket itself, so that the cylinder was supplied with pure, dry, adheated steam alone, and was, moreover, kept surrounded with steam of nearly the boiler pressure, which, being much greater than the cylinder pressure, caused the cylinder and valve-chest surfaces to become an adheater as well as the surface of the coil.

Adheated in the Tubular Adheater. After the completion of the experiments with the steam adheated in the coil, and in view of the great economic gain produced by it, it was considered desirable to push the experiments still further in the same direction, with a considerable addition of adheating surface,

for the purpose of ascertaining whether the limit of gain by this process of adheating had been reached in the coil. As the coil occupied all the space between the cylinder and jacket, any further extension of it was impossible; and it was decided to construct an adheater in the form of a separate vessel outside of the jacket, into which any required area of adheating surface might be placed. This arrangement was the more desirable, as in the case of large engines it was what would probably have to be adopted, and it was, of course, better that the experiment should be made as nearly as possible under the conditions that would obtain in actual practice. Accordingly, a tubular adheater was constructed consisting of a cylindrical shell with flat ends. These ends were hollow, in order that the included space might be filled with boiler steam, so that the inner area of each end should be steam-jacketed. The inner side of each hollow end was drilled and used for a tube-plate; and these plates were connected by tubes which formed the adheating surface. The axis of the cylindrical shell was placed horizontally, which was also the position of the tubes within. By this arrangement the adheater was divided into an exterior and an interior compartment, the first being composed of the space between the cylindrical part of the shell, the tube-plates, and the outside of the tubes, the last being the space within the tubes, and between the tube-plates and ends of the shell. The steam-pipe was connected upon the top and near one end of the shell. At the opposite end and also upon the top, a short elbow-pipe, containing the throttle-valve, connected the cylindrical part of the shell with one of the ends. The opposite end was connected with the cylinder valve-chest by a steam-jacketed pipe: the steam to fill this jacket and the adjacent hollow end was taken from the cylinder-jacket, the communication between the three being open.

With this arrangement the steam from the boiler was delivered into the exterior compartment of the adheater, which was thus made to act as a priming-box. From this compartment it passed through the elbow-pipe into the interior compartment, undergoing throttling to any required degree at the throttle-valve, and entering the interior compartment at a less pressure and temperature than it had in the exterior compartment. From the interior compartment it passed freely to the cylinder valve-chest. By means of this system the steam in the interior compartment could be adheated to a very considerable degree by the steam in the exterior compartment, depending on the amount of throttling given.

The cylinder-jacket was filled through a small branch of the main steam-pipe with steam of nearly boiler pressure, so that the steam, from the throttle-valve to the end of the stroke of the piston, was not only completely prevented from losing any heat by radiation or conduction, but it was constantly exposed to the action of steam of much higher temperature, whereby the temperature due to its pressure as saturated steam was considerably increased.

All the water resulting from the condensation of steam in the steam-pipe, or passed over from the boiler in the liquid state, was deposited in the cylindrical part of the shell and drawn off continuously with the water of condensation due to the external surfaces of the adheater. Consequently, no water could enter the cylinder which was thus supplied with pure, dry, adheated steam. The water of condensation in the jacket of the pipe connecting the adheater and valve-chest, fell into the cylinder-jacket and was drawn off continuously with the water of condensation from that jacket.

After the construction of the tubular adheater, the coil was permanently removed from the cylinder-jacket.

The adheater was placed in a wooden box and surrounded with sawdust six inches thick as a non-conductor. All the connecting pipes were thoroughly felted.

Adheated in the Steam-jacketed Steam-pipe. The results from the tubular adheater proving less instead of more than from the coil, and as it appeared that all the advantages of the system of adheating

could be derived from a moderate area of surface, another and simpler method was essayed for effecting it. This consisted in converting the surface of the steam-pipe into adheating surface by steam-jacketing it. To give effect to this idea the steam-pipe from the boiler to the cylinder-jacket was enclosed in another pipe, and the annular space between them filled with boiler steam. The throttle-valve was now placed at the boiler end of the steam-pipe. The steam was delivered direct from the boiler into the cylinder valve-chest; and to intercept any water that might pass over from the boiler in the liquid state, a priming-box was interposed in the steam-pipe between the boiler and throttle-valve, so that the cylinder should be supplied with pure, dry adheated steam alone. The cylinder-jacket was filled with steam through a small branch of the main steam-pipe. The exterior of the jacketed pipe was thoroughly protected by a thick covering of felt The water resulting from the condensation of steam in the jacket of the steam-pipe flowed into the cylinder-jacket, and was continuously drawn off with the water of condensation due to that jacket.

From this description it will be perceived that, by the simple action of the throttle-valve, the pressure of the steam within the entire length of the steam-pipe could be made less to any required degree than the pressure of the steam in the surrounding annular space, and that its temperature due to its pressure as saturated steam would thus be increased by the greater temperature of the surrounding steam.

In the experiments recorded in columns O, P, S, T, U, and X, the steam was cut off in the cylinder at 0·19 of the stroke of the piston from the commencement; and in the remaining columns Q, R, V, W, and Y, it was cut off at 0·83 of the stroke of the piston from the commencement.

In experiments O, P, S, and V, the diameter of the engine shaft pulley was 30 inches. In experiment Q it was 60 inches. In experiments R, T, W, X, and Y, it was 36 inches. And in experiment U it was 24 inches. The diameter of the fan shaft pulley was $9\frac{5}{8}$ inches in all.

## CONTENTS OF THE LINES.

Total Quantities. Line 1 contains the duration of the experiments. Experiments B, K, N, O, Q, and S, were each of 60 consecutive hours. Columns A, L, and P, contain two experiments, each of 60 consecutive hours. Experiments C, D, F, G, H, J, M, T, U, X, and Y, were each of 30 consecutive hours. Columns E, I, V, and W, contain two experiments, each of 30 consecutive hours. Column R contains three experiments, one of 17 consecutive hours, and two of 15 consecutive hours.

Line 2 contains the total number of double strokes of the engine piston made during the experiments, as given by the counter.

Line 3 contains the total number of revolutions made by the fan. They are the quantities on line 2 increased in the ratio of $9\frac{5}{8}$ to the diameters of the pulleys given in the headings of the columns.

Line 4 contains the total number of pounds of feed-water pumped into the boiler, as determined by the capacity of the feed-water tank and weight per cubic foot at the temperature on line 30.

Line 5 contains the total number of pounds of water drawn off continuously from the cylinder-jacket and accurately weighed. In experiments K, L, O, P, Q, and R, this weigh tincludes the water resulting from the condensation of steam in the steam-pipe and cylinder-jacket, and also the water passed over from the boiler in the liquid state, if there were any. In experiments M, N, X, and Y, it includes only the water resulting from the condensation of steam in the cylinder-jacket and small branch of the main steam-pipe through which that jacket was filled with steam. In experiments S, T, U, V, and W, it includes the water resulting from the condensation of steam in the cylinder-jacket, in the small branch of the main steam-pipe through which that jacket was filled with steam, and in the jacket of the short pipe connecting the tubular adheater with the cylinder valve-chest.

Line 6 contains the total number of pounds of water drawn off continuously from the tubular adheater and accurately weighed. This weight includes the water resulting from the condensation of steam in the steam-pipe and adheater, and also the water passed over from the boiler in the liquid state, if there were any.

Line 7 contains the total number of pounds of injection-water and condensed steam discharged from the condenser by the air-pump, as determined by the capacity of the tank for receiving the same, and weight per cubic foot at the temperature on line 28.

Line 8 contains the total number of pounds of injection-water and condensed steam that would have been discharged from the condenser by the air-pump, had the weight of the steam condensed been only that which was discharged from the cylinder at the end of the stroke of the piston, as given on line 52, the temperature of the injection-water being that given on line 27, the temperature of the hot-well that given on line 28, and the total heat of the steam that which is due to steam of the final pressure in the cylinder, as given on line 37. The calculations were made by the following formula, in which let S = the number of pounds of steam discharged from the cylinder (line 52), $h$ = the total heat of the same, $t$ = the temperature of the hot-well, $i$ = the temperature of the injection-water, and W = the number of pounds of injection-water and condensed steam discharged; then $W = S \times \frac{h - t}{t - i} + S$.

Line 9 contains the total number of pounds of injection-water and condensed steam that would have been discharged from the condenser by the air-pump, had the weight of steam condensed been the weight of feed-water pumped into the boiler (line 4) less the weight of water drawn off from the cylinder-jacket (line 5) and tubular adheater (line 6), the total heat of this steam being that due to the final pressure in the cylinder (line 37), and the temperatures of the injection-water and hot-well, those given on lines 27 and 28. The calculations were made by the above formula.

Line 10 contains the number of pounds of Trevorton semi-anthracite consumed, accurately weighed.

Line 11 contains the total number of pounds of refuse in ashes, clinker, and fine coal given by the Trevorton coal. It was carefully weighed in the dry state.

Line 12 contains the total number of pounds of combustible consumed. They are the quantities on line 11 less those on line 12.

Line 13. The quantities on this line are the per centum which the quantities on line 11 are of those on line 10.

QUANTITIES PER HOUR. The quantities on lines 14, 15, 16, 17, 18, and 19 are respectively the quotients of those on lines 4, 5, 6, 7, 10, and 12, divided by the number of hours on line 1.

The quantities on lines 20 and 21 are respectively the quotients of those on lines 10 and 12, divided by the number of hours on line 1, and by the number of square feet of grate surface in the boiler.

ENGINE. The quantities on lines 22 and 23 are the quotients of the quantities on lines 2 and 3, divided by the number of minutes on line 1.

Line 24 contains the average height of the barometer in inches, the observations being noted every hour during the experiments. The barometer was situated about ten feet from the engine, and its indications are the value of the atmospheric pressure.

Line 25 contains the average vacuum in inches of mercury in the condenser by an open mercurial gauge; the observations were noted every hour. The vacuum thus expressed is, of course, the difference between the atmospheric pressure and the pressure in the condenser.

TEMPERATURES. The temperatures contained in lines 26, 27, 28, 29, 30, and 31 are expressed in degrees

Fahrenheit. The same thermometers were employed in the same positions throughout, and their indications corrected by a common standard. The observations were noted hourly, and the quantities given are the averages.

Line 26 contains the temperature in the cylinder-jacket. It was that of the steam when steam was admitted, and that of the air when steam was excluded. The bulb and a large portion of the stem of the thermometer was exposed directly to the temperature of the steam or air.

Line 27 contains the temperature of the injection-water as it entered the condenser. Care was taken to obtain it from a pipe outside the engine room, constantly used, and exposed to the temperature of the external atmosphere.

Line 28 contains the temperature of the mixture of injection-water and condensed steam discharged from the condenser by the air-pump. The pipe leading from the top of the pump to the receiving tank was about three feet long, and the thermometer was permanently situated at the mouth of the pipe, so as to be constantly covered by the flow of the discharge. Repeated trials showed no difference in the indication when the thermometer was placed in the top of the pump immediately over the delivery-valve.

Line 29 contains the temperature of the adheated steam. It is given in the case of the tubular adheater only. The bulb of the thermometer was placed in the current of adheated steam, leaving the adheater for the cylinder valve-chest, and was, together with the immersed part of the stem, carefully shielded from the radiation of the surfaces exposed to the temperature of the boiler steam. In the case of adheating by the coil, and by the steam-jacketed steam-pipe, it was found impossible to so shield the thermometer from the effects of radiation from the surfaces exposed to the temperature of the boiler steam as to make its indications reliable.

Line 30 contains the temperature of the feed-water, as given by a thermometer kept constantly immersed in the tank.

Line 31 contains the temperature of the engine room. The thermometer hung midway between the boiler and engine, and was sheltered from the direct radiation of both.

Steam Pressures. Lines 32, 33, and 34 contain, respectively, the steam pressures in pounds per square inch *above the atmosphere*, in the boiler, in the cylinder-jacket, and in the cylinder valve-chest. These pressures are by large and fine manometer gauges manufactured by the "*Novelty Iron Works.*" Their accuracy was frequently tested. The observations were noted every hour, and the quantities given are the averages. These pressures varied but little during an experiment, the steam pressure in both boiler and cylinder being very uniformly maintained by the pressure damper for the one, and by the governor for the other. The *total* or true pressure of the steam in the boiler, cylinder-jacket, and cylinder valve-chest,—that is to say,—its pressure above zero,—will be obtained by adding the pressures per gauge to the atmospheric pressure, as given on line 24.

Lines 35, 36, 37, and 38 contain the steam pressures in the cylinder, as given by the indicator. A diagram was taken every hour from each end of the cylinder, and the quantities in the above lines are the means of all. The diagrams varied very little during each experiment, owing to the uniformity in the supply of the steam, and in the resistance of the load; and it is believed that the mean of so large a number, taken at regular intervals by an excellent instrument, may be relied on with great certainty. The steel spring of the indicator being graduated from the atmospheric line, the total or true steam pressures given in lines 35, 36, and 37, are those due to the atmospheric pressure, as shown by the barometer on line 24,—that is to say, these pressures are all given above the absolute zero. The pressure on line 39 is given correctly by the instrument, as it is not affected by atmospheric variation.

POWER. *Absolute.* Lines 40, 41, and 42 contain the power developed by the engine expressed in horses. This dynamical unit is the habitual one of 33,000 pounds raised one foot high per minute. The speed of the piston is obtained from line 22, and in the calculation its net area—that is exclusive of its rod—is employed.

On line 40 is the gross effective horse power, calculated for the pressure on line 39. It is inclusive of the power required to work the engine *per se*, and of the power required to overcome the friction and resistance of the load, but exclusive of the power required to overcome the back pressure (line 38).

On line 41 is the total horse power, calculated for the sum of the pressures on lines 38 and 39. It includes the power required to work the engine *per se*, the power required to overcome the friction and resistance of the load, and the power required to overcome the back pressure. It is the entire dynamical effect produced by the steam without reference to its utility.

On line 42 is the net horse power, calculated for the pressure on line 39 less $3\frac{1}{2}$ pounds per square inch of piston, which is deducted for the pressure required to work the engine *per se*. It is the power required to overcome only the friction and resistance of the load, and expresses the commercial dynamical effect,—that is, the power usefully employed in moving the load. The pressure required to work the engine *per se*, with the fan disconnected, was found by numerous trials made at intervals in the course of the experiments to average $3\frac{1}{2}$ pounds per square inch of piston by indicator.

*Economic.* Lines 43, 44, and 45 contain, respectively, the number of pounds of feed-water consumed per hour per gross effective, total, and net indicated horse power. These weights are the quotients of the quantity on line 14, divided by the quantities on lines 40, 41, and 42.

Lines 46, 47, and 48 contain, respectively, the number of pounds of coal consumed per hour per gross effective, total, and net indicated horse power. These weights are the quotients of the quantity on line 18, divided by the quantities on lines 40, 41, and 42.

Lines 49, 50, and 51 contain, respectively, the number of pounds of combustible consumed per hour per gross effective, total, and net horse power. These weights are the quotients of the quantity on line 19, divided by the quantities on lines 40, 41, and 42.

TOTAL EVAPORATION. Line 52 contains the total number of pounds of steam discharged from the cylinder at the end of the stroke of the piston. It is calculated from the weight per cubic foot of the steam of the pressure at the end of the stroke (line 37), as determined by FAIRBAIRN'S formula, from the total number of double strokes of piston made (line 2), and from the space displacement of the piston per stroke, exclusive of rod, plus the space comprised in the clearance and steam passages at one end of the cylinder, due allowance having been made for the back pressure (line 38) already occupying this space when the valve commences to open the steam port. The weight of steam thus calculated was evaporated from the temperature of the feed-water (line 30).

Line 53 contains the total number of pounds of steam condensed in the cylinder to furnish the heat transmuted into the total power developed by the engine, according to JOULE'S equivalent of one pound of water raised one degree of temperature on Fahrenheit's scale for every 772 foot-pounds developed by the engine, which would make the thermal equivalent of one indicated horse power $\left(\frac{33,000}{772} =\right) 42{\cdot}7461$ pounds of water raised one degree Fahr.

The quantities were calculated by the following formula: Let

$k =$ the number of total indicated horses power developed by the engine (line 44).

$e =$ the total heat of steam of the boiler pressure in degrees Fahr. (line 32).

$g$ = the temperature in degrees Fahr. of the feed-water from which the evaporation is derived (line 30).

$t$ = the time in minutes during which the total indicated horses power acted (line 1).

Then $\frac{k \times 42{\cdot}7461 \times t}{e - g}$ = the number of pounds of steam that would be evaporated in the time $t$, from a temperature $g$ of feed-water, by the heat transmuted into the total indicated horses power $k$ developed by the engine (line 53).

Line 54 contains the total number of pounds of steam condensed in the cylinder-jacket, steam-pipe, and tubular adheater, by external radiation and by adheating the steam. It is the sum of the quantities on lines 5 and 6, and was evaporated from the temperature of the feed-water (line 30).

Line 55 contains the total number of pounds of steam that would have been evaporated had the temperature of the feed-water been 100° Fahr., according to the indicator determination, plus the number of pounds of steam condensed in the cylinder-jacket, steam-pipe, and tubular adheater. It is the sum of the quantities on lines 52, 53, and 54, corrected for the difference of temperature of feed-water 100° Fahr., and of the temperatures on line 30.

Line 56 contains the total number of pounds of steam that would have been evaporated had the temperature of the feed-water been 100° Fahr., according to the tank determination. It is the quantity on line 4, corrected for the difference of temperature of feed-water 100° Fahr., and of the temperatures on line 30.

Line 57 contains the total number of pounds of steam that would have been evaporated had the temperature of the feed-water been 212° Fahr., according to the indicator determination, plus the number of pounds of steam condensed in the cylinder-jacket, steam-pipe, and tubular adheater. It is the sum of the quantities on lines 52, 53, and 54, corrected for the difference of temperature of feed-water 212° Fahr., and of the temperature on line 30.

Line 58 contains the total quantity of pounds of steam that would have been evaporated had the temperature of the feed-water been 212° Fahr., according to the tank determination. It is the quantity on line 4, corrected for the difference of temperature of feed-water 212° Fahr., and of the temperature on line 30.

Economic Evaporation. Lines 59 to 66, both inclusive, contain the number of pounds of steam evaporated per pound of coal and per pound of combustible, from the temperatures 100° and 212° Fahr. of feed-water, according to both tank and indicator determinations, including in the latter the steam condensed in the cylinder-jacket, steam-pipe, and tubular adheater. The evaporation is given from both temperatures for convenience of comparison.

Lines 59 and 63 contain the number of pounds of steam evaporated per pound of coal and per pound of combustible, from the temperature 100° Fahr. of feed-water, according to the determination by the indicator, &c. They are the quotients of the quantity on line 55, divided by the quantities on lines 10 and 12 respectively.

Lines 60 and 64 contain the number of pounds of steam evaporated per pound of coal and per pound of combustible, from the temperature 100° Fahr. of feed-water, according to the determination by the tank. They are the quotients of the quantity on line 56, divided by the quantities on lines 10 and 12 respectively.

Lines 61 and 65 contain the number of pounds of steam evaporated per pound of coal and per pound of combustible, from the temperature 212° Fahr. of feed-water, according to the determination by the indicator, &c. They are the quotients of the quantity on line 57, divided by the quantities on lines 10 and 12 respectively.

Lines 62 and 66 contain the number of pounds of steam evaporated per pound of coal and per pound of combustible, from the temperature 212° Fahr. of feed-water, according to the determination by the tank. They are the quotients of the quantity on line 58, divided by the quantities on lines 10 and 12 respectively.

## TABLE No. 2.

The quantities given in Table No. 1 are exactly those which were obtained under the experimental conditions. Two of these conditions, however, which ought to have been constant, were variable by circumstances beyond control, namely, the friction of the engine and the back pressure vapor against its piston. The variations in the first were very slight, but in the second they were considerable. Now, as these variations have no connexion with the method of using steam in the different experiments, and, being purely accidental, might have been reversed, it is necessary to eliminate them, in order to arrive at true comparative results. This has been done in Table No. 2, and to render the results more generally applicable, the pressure required to work the engine *per se* has been assumed at 2 pounds per square inch of piston—the average for medium sized condensing engines—instead of the 3½ pounds per square inch which was required for the experimental engine. The back pressure against the piston has been taken in all cases, except for the experiments in columns Z and A, at 2½ pounds per square inch, which may be regarded as the minimum in the best practice. With these corrections the quantities in Table No. 2 are a selection of such of those in Table No. 1 as are necessary, with others deduced from them, to effect a proper and complete comparison between the results given by the different experiments. In this table, as in Table No. 1, the quantities are grouped, the columns lettered, and the lines numbered for facility of reference.

STEAM PRESSURES. Lines 1, 2 and 3 contain the absolute pressures—that is, the total pressure above zero—in pounds per square inch of the steam in the boiler, in the cylinder-jacket, and in the cylinder valve-chest, as given by manometer gauges which were carefully tested and corrected for each experiment. These gauges gave the pressure above the atmosphere (lines 32, 33, and 34, Table No. 1), to which was added the pressures due to the height of the barometer (line 24, Table No. 1), in order to obtain the above absolute or total pressures.

Lines 4 to 12, both inclusive, contain the steam pressures in the cylinder in pounds per square inch by indicator.

Line 4 contains the absolute pressure at the commencement of the stroke of the piston (line 35, Table No. 1).

Line 5 contains the absolute pressure at the point of cutting off the steam (line 36, Table No. 1).

Line 6 contains the absolute pressure at the end of the stroke of the piston (line 37, Table No. 1).

Line 9 contains the mean total pressure during the stroke of the piston—that is to say, the average pressure above zero. It is the sum of the quantities on lines 38 and 39 of Table No. 1. The zero for the quantities on lines 4, 5, 6, and 9 was obtained from the height of the barometer on line 24 of Table No. 1.

Line 7 contains the assumed constant back pressure against the piston above zero; it is 2½ pounds per square inch above zero, instead of the quantities on line 38, Table No. 1.

The quantities on line 8 are the remainders of those on line 9, after deducting from them the constant back pressure of 2½ pounds per square inch, as given on line 7. The quantities on line 8 are the mean gross effective pressures on the piston in pounds per square inch, and correspond to what is commonly termed the average indicator pressure.

Line 10 contains the assumed constant pressure required to work the engine *per se*. It is 2 pounds per square inch of piston.

The quantities on line 11 are the remainders of those on line 8, after deducting those on line 10. They

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are the mean net pressures on the piston in pounds per square inch—that is to say, the pressure usefully applied to the work which remains after deducting from the total pressure (line 9) the sum of the friction and back pressures (lines 7 and 10).

Line 12 is the per centum which the quantities on line 11 are of those on line 9. They show, comparatively, the proportion of the total pressure which is usefully applied in the various experiments.

TEMPERATURES. Lines 13 to 18, both inclusive, contain the temperatures in degrees Fahrenheit necessary to be known in investigating the results of the experiments.

On line 13 is the temperature of the engine and boiler room. It is line 31 of Table No. 1.

On line 14 is the temperature of the air or steam in the cylinder-jacket, according as the experiment was made with or without steam in the jacket. It is line 26 of Table No. 1.

In observing the temperature when the jacket was filled with air, it must be remembered that the cylinder valve-chest was immersed in this jacket as well as the cylinder, and, being constantly filled with steam of the pressure on line 3, greatly increased the temperature above what it would have been had the cylinder alone been immersed. There was no circulation of air through this jacket, the only aperture being a hole on the top two inches in diameter, in which the thermometer was inserted.

The temperature of the steam in the jacket was taken by a mercurial thermometer whose indications did not vary by more than three degrees from that due to the pressure shown by the gauge, as given in the table published in the article Steam of the last edition of the Encyclopedia Britannica. As the gauge was doubtless more accurate than the thermometer, I have substituted the temperatures of the table for the observed ones. The bulb and part of the stem of the thermometer was immersed; its indications varied from two to four degrees from those of the table, and were always higher.

On line 15 is the average temperature of the steam as saturated steam on the steam side of the piston during a stroke. It is the temperature corresponding to the pressure on line 9, as given by the table just referred to.

On line 16 is the temperature of the back pressure vapor considered as saturated. It is the temperature corresponding to the pressure on line 38 of Table No. 1, as given by the Table in the Encyclopedia.

On line 17 is the average temperature of the steam in the cylinder on both sides of the piston during a stroke considered as saturated. It is the mean of the temperatures on lines 15 and 16.

On line 18 is the temperature of the adheated steam in the interior compartment of the tubular adheater. It was taken from a tested mercurial thermometer, the bulb and part of the stem of which was immersed in the centre of the steam current as it passed out of the adheater. The difference between the temperature on this line and that due to the pressure on line 3, show the degree to which the steam was adheated nearly.

POWER. *Absolute.* Lines 19, 20, and 21 contain the absolute indicated horse power developed by the engine. The horse power being the usual 33,000 pounds raised one foot high per minute.

On line 19 is the *gross effective* horse power. It is calculated for the pressure on line 8, for the area 21·15 square inches of the piston, and for the speed of piston on line 22 of Table No. 1. It is what is usually known as the indicated horse power.

On line 20 is the *total* horse power. It is calculated for the pressure on line 9, and for the area and speed of piston as above. It is the quantity on line 41, Table No. 1, and is the expression for the total dynamic effect produced by the steam which entered the cylinder, and includes the useful work, and overcoming the back pressure and friction resistances.

On line 21 is the *net* horse power. It is calculated for the pressure on line 11, and for the area and

speed of piston as above. It is the expression for only that part of the total power developed by the engine which is applied to useful work, being exclusive of that which was consumed in overcoming the back pressure and friction resistances.

*Economic.* Lines 22 to 27, both inclusive, contain the cost of the gross effective, total, and net indicated horse power in pounds per hour of steam furnished from the boiler under two conditions, namely:—1st. *Inclusive* of the weight of steam condensed in the cylinder-jacket, steam-pipe, and tubular adheater, which gives the *gross* cost of producing the different powers in pounds of the total amount of steam evaporated in the boiler. 2d. *Exclusive* of the weight of steam condensed in the cylinder-jacket, steam-pipe, and tubular adheater, which gives the *net* cost of producing the different powers in pounds of steam that entered the cylinder, without reference to the total pounds required to be evaporated in the boiler to furnish it.

The quantities on lines 22, 23, and 24 are the quotients resulting, respectively, from the division of the quantities on line 14 of Table No. 1 by those on lines 19, 20, and 21.

The quantities on lines 25, 26, and 27 are the quotients resulting, respectively, from the division of the quantities on line 14, less those on lines 15 and 16 of Table No. 1, by the quantities on lines 19, 20, and 21.

CONDENSATION. Lines 28 to 33, both inclusive, contain the condensation of steam, expressed proportionally, that occurred in the cylinder, and in the cylinder-jacket, steam-pipe, and tubular adheater under the various conditions of the experiments.

Line 28 contains the condensation of steam in the cylinder due to the development of the total power (line 20), expressed in per centum of the weight of steam evaporated in the boiler. It is the per centum which the quantities on line 53, Table No. 1, are of those on line 4, Table No. 1.

Line 29 contains the aggregate condensation of steam in the cylinder-jacket, steam-pipe, and tubular adheater, expressed in per centum of the weight of steam evaporated in the boiler. It is the per centum which the quantities on lines 5 and 6, Table No. 1, are of those on line 4, Table No. 1. This condensation is the sum of that which is due to external refrigeration, and to adheating the steam used in the cylinder.

Line 30 contains the condensation of steam in the cylinder, exclusive of that (line 28) which resulted from the development of the power, expressed in per centum of the weight of steam evaporated in the boiler. If we deduct from the weight of steam evaporated in the boiler (line 4, Table No. 1), the weight condensed in the cylinder-jacket, steam-pipe, and tubular adheater (lines 5 and 6, Table No. 1), the weight condensed in the cylinder to produce the power (line 53, Table No. 1), and the weight discharged from the cylinder at the end of the stroke of its piston (line 52, Table No. 1), then the quantities on line 30 are the per centum which the remainders are of the quantities on line 4, Table No. 1.

Line 31 contains the condensation of steam in the cylinder due to the development of the total power (line 20), expressed in per centum of the remainder of the weight of steam evaporated in the boiler after deducting the weight condensed in the cylinder-jacket, steam-pipe, and tubular adheater. It expresses the condensation due to the rendition of the power in per centum of the steam which actually entered the cylinder. It is the per centum which the quantities on line 53, Table No. 1, are of the quantities on line 4, Table No. 1, less the sum of the quantities on lines 5 and 6, Table No. 1.

Line 32 contains the condensation of steam in the cylinder, exclusive of that which resulted from the development of the power, expressed in per centum of the weight of steam which actually entered the cylinder. If from the total weight of steam evaporated in the boiler (line 4, Table No. 1) there be deducted the weight condensed in the cylinder-jacket, steam-pipe, and tubular adheater (lines 5 and 6, Table No. 1), there will be a certain remainder which is the weight of steam that actually entered the cylinder: call this

weight A. If from this remainder there be deducted the quantities on line 53, Table No. 1, which are the weights condensed to produce the power, and the quantities on line 52, Table No. 1, which are the weights discharged from the cylinder at the end of the stroke of its piston, there will still remain certain weights, and the quantities on line 32 are the per centum which these weights are of the quantities A.

Line 33 contains the total condensation, due to all causes, effected upon the weight of steam evaporated in the boiler, expressed in per centum of that weight. If from the total weight of steam evaporated in the boiler (line 4, Table No. 1) there be deducted the weight discharged from the cylinder at the end of the stroke of its piston (line 52, Table No. 1), there will be a certain remainder, and the quantities on line 33 are the per centum which this remainder is of the quantities on line 4, Table No. 1.

INJECTION-WATER. *Absolute.* Line 34 contains the number of pounds of injection-water that would have been required per hour to condense the steam which actually entered the cylinder, supposing the whole of it to have been discharged into the condenser at the end of the stroke of the piston, and at the pressure in the cylinder at the end of the stroke (line 6). It is obtained by subtracting the quantities on line 4, Table No. 1, from those on line 9, Table No. 1, and dividing the remainder by those on line 1, Table No. 1. This supposes no water to have passed from the cylinder to the condenser.

Line 35 contains the pounds of injection-water that would have been required per hour to condense the steam, supposing that no more was discharged from the cylinder than the number of double cylinderfulls (line 2 divided by line 1, Table No. 1) of the pressure at the end of the stroke of the piston, line 6. It is obtained by subtracting the quantities on line 53, Table No. 1, from those on line 8, Table No. 1, and dividing the remainder by those on line 1, Table No. 1. This supposes that no more steam entered the cylinder than was discharged from it, as measured by the number of cylinderfulls of the pressure at the end of the stroke of the piston.

Line 36 contains the pounds of injection-water actually used per hour to condense the steam which passed from the cylinder to the condenser during the whole time the exhaust passage was open. It is obtained by subtracting the quantities on line 4, Table No. 1, from those on line 7, Table No. 1, and dividing the remainder by those on line 1, Table No. 1.

*Proportional.* Line 37 contains the per centum which the quantities on line 35 are of those on line 34. It exhibits the enormous difference that exists in all cases between the weight of injection-water required, calculated from the pressure of the steam in the cylinder at the end of the stroke of the piston, and calculated from the weight of feed-water pumped into the boiler.

Line 38 contains the per centum which the quantities on line 36 are of the quantities on line 34. It exhibits the difference that exists, in all cases, between the weight of injection-water actually used, by measurement, and the weight required, by calculation, from the weight of feed-water pumped into the boiler.

## DISCUSSION OF THE RESULTS.

The steam-engine is simply an assemblage of organs, by means of which the pressure (a statical effect) of steam is made to develop a dynamic effect, easily controllable by man, and generally applicable to any purpose requiring the agency of power. It is not a machine, because its sole use is the production (not creation) of power, whereas the function of a machine is the application of power to the production of some specific effect or change upon matter. The one is universal and operative of all machines; the other is limited to the application of power to effect one specific thing in one particular way. The pressure of steam is the result of an unknown force, and depends for its intensity on the resistance, either statical or dynamical,

opposed to it. Pressure and power, then, are the manifestations upon matter of an unknown and elemental force. They are the same in one point of view, for power may be considered as quantity of pressure, being, for equal pressures, in the direct ratio of the space over which the pressure moves.

The economic problem in connexion with steam is the quantity of dynamic effect obtained from a given weight; and it is affected both by the mechanical conditions of the engine and by the manner in which the steam is used, the results varying widely with variations of these conditions.

In the absence of knowledge, *a priori*, regarding the influences exercised by different sizes and types of engines, and by the proportions and kind of their organs, and in view of our knowledge of only some of the physical laws of steam, which, preventing the formation of a complete theory of its mode of action, renders impossible precise predictions of results due to different manners of using it, and in different engines, we are compelled to rely wholly on experiment as an oracle, and to consult it in each particular case. The object of such experiments should be the ascertaining of the cost of any dynamic unit—the horse power of WATT for convenience—in weight of steam. The common practice is to ascertain it in weight of fuel consumed, but this method, though convenient, is too rude to give other than roughly approximative results, as it is vitiated by all the circumstances connected with the generation of the steam, such as quality of coal, evaporative efficiency of boiler, skill of firemen, barometric and hygrometric conditions of the atmosphere, &c. In any experiments, then, pretending to accuracy, the cost of the power must be ascertained in weight of steam alone. In extended experiments embracing very long times, and when the object is only comparative results, the weight of coal may be used, but even then there is always wanting the *certainty* that attends the steam measure. The coal measure is only proper when two entire systems are to be compared, including both the generation and the use of the steam.

In the following discussion of the experiments, the dynamic unit employed is the horse power of 33,000 pounds raised one foot high per minute, as determined by the indicator. Now, were it not for the imperfect manner in which steam is used in the steam-engine, the only power measured by the indicator would be the total horse power,—that is, the power calculated for the steam pressure from zero, because there would be neither resistance of back pressure in the cylinder resulting from imperfect condensation, or the presence of air, nor resistance of mechanism resulting from its friction; but as both exist and require *pro rata* power to overcome them as much as the visible external load does, there arise, in addition to the total horse power, the gross effective and the net horse power, both of which, and especially the latter, is of the utmost importance in the practical commercial treatment of the problem.

The total horse power is the expression for the entire dynamic effect produced by the steam that enters the cylinder, irrespective and inclusive of the back pressure and friction resistances, and is, therefore, the measure of the whole work done or effect produced, and is to be employed in comparing the general or abstract efficiency of different manners of using steam. The gross effective horse power, which is the total horse power less the horse power required to overcome the back pressure resistance, measures the economic efficiency of the engine itself in function of the proportions and kind of its organs. It is, therefore, useful in comparing the relative excellence of such organs and their proportions, but gives no correct answer to any question bearing upon the different manners of using steam. The net horse power is what remains of the total power after deducting for both the back pressure and the friction resistances. It is the power applied usefully to the external load or visible work, and is the true commercial unit, because it alone produces the effects desired and intended, whereas the other effects of overcoming back pressure and friction resistances are neither desired nor intended, but were unavoidable from the constitution of matter. In this, as in all other things, we can never do what we wish without doing also what we do not wish, the phenomena

of nature being so interlocked that no purely single and separate effect can be produced. The back pressure and friction resistances, though inseparable from the use of steam, may vary widely with different proportions and types of engines, and with accidental circumstances such as air-leaks, lubrication, &c.; but though they cannot be avoided they can be reduced to a minimum which can be made the same in all cases, for it is obvious that it is possible to so proportion all engines as to obtain the same condensation, and, with engines of the same capacity of cylinder, the same friction resistance per stroke of piston. Experiment tells us what this minimum is, and in comparing the results of experiments it should be assumed in all cases, and the gross effective and net horses power deduced from the total power in accordance with it. This is what has been done in Table No. 2. I shall, therefore, in the following discussion, use the total horse power in determining the general or abstract efficiency of the steam as employed in the different manners of the experiments, and the net horse power in determining the practical or commercial efficiency of the steam similarly used. The gross effective horse power, which determines only the efficiency of the engine in function of its organs, is of no use in this discussion and will not be further referred to. It is included in Table No. 2, because being generally, though improperly, employed as the correct measure of the dynamic effect produced by the steam; it may be welcome to some readers from their familiarity with it.

I shall, then, discuss the problem of the economic efficiency of steam used in the different manners and under the different conditions of the experiments, by means of the pounds of steam required per hour in each case to produce the total and the net horse power (lines 23, 24, 26, and 27, Table No. 2). If any one, more familiar with fuel measurement, can give greater precision to his ideas by taking the pounds of coal or of combustible consumed per hour per indicated horse power, for the measure of the economic efficiency, he has only to divide the pounds of steam by the number which he assumes to be the pounds of steam evaporated per pound of fuel.

The experiments comprised the following manners of using steam, namely:—1st. "Saturated Steam used with Air in the Jacket." 2d. "Adheated Steam used with Air in the Jacket." 3d. "Saturated Steam used with Steam in the Jacket." 4th. "Adheated Steam used with Steam in the Jacket," the adheating being effected in different experiments by the *coil*, by the *tubular adheater*, and by the *steam-jacketed steam-pipe*, with a view to determine the relative efficiency of these three instruments. The conditions of the experiments were greatly varied by using the steam with both a high and a low measure of expansion, and with widely varying boiler and cylinder pressures. The results will be discussed separately, and first

OF SATURATED STEAM USED WITH AIR IN THE JACKET. Columns A to H, both inclusive, contain the experiments made with saturated steam, the cylinder-jacket being filled with air. Of these experiments, those in columns A, B, C, and D were made with the steam cut off at 0·19 of the stroke of the piston from the commencement, while those in columns E, F, G, and H were made with the steam cut off at 0·83 of the stroke of the piston from the commencement. We have consequently the steam used with both a high and a low measure of expansion. Further, the conditions were varied by having, during the experiments recorded in columns B D, and F, the cylinder inside the jacket covered with felt two inches thick, while during the remaining experiments this covering was removed. In both cases, however, the jacket was well felted. The boiler and cylinder pressures were greatly varied in the different experiments.

We have, then, in these experiments to examine—

1st. The absolute economic efficiency of the steam—that is, the cost of the total and net horse power in pounds of steam.

2d. The effect of the different pressures in the cylinder.

3d. The economic gain due to the higher measure of expansion. And

4th. The influence of the felting on the cylinder within the jacket.

1st. In observing the *cost of the total and net horse powers* (lines 23 and 24, Table No. 2), we are struck with the enormous weight of steam expended, which is about two-and-a-half times as much as is habitually required with medium sized engines under good working conditions, but on referring to line 30 of Table No. 2 the cause becomes apparent, for we there find that, exclusive of the condensation due to the production of the power (line 28, Table No. 2), there was condensed in the cylinder nearly 60 per centum of all the steam generated in the boiler. In examining this condensation more closely we find, too, that it was apparently nearly as great when the steam was cut off at 0·83 of the stroke of the piston as when the cutting off was done at 0·19 of the stroke,—another wide departure from general experience. What could have caused these anomalous results? Now, as there was neither steam leakage nor priming, they could not be due to either; and as the cylinder was felted inside the jacket, between which felt and the metal of the jacket there intervened an air space followed by the metal of the jacket and its felting, and as the temperature of the room averaged about 82° Fahr., they clearly could not be due to external refrigeration from the cylinder alone. The only remaining cause is the refrigerating effect of the temperature of the back pressure vapor upon the interior surfaces of the cylinder. But why should this refrigerating effect be so much greater in our small experimental cylinder than is found in the larger ones of every day practice? The answer lies mainly in that very smallness of the cylinder, for when we consider its dimensions we perceive at once that, in comparison of its capacity, the development of interior surface is enormous. In addition to the interior surfaces proper—that is to say, the cylindrical surface, the surface of the head, of the piston disc, of the steam passage, and of the under side of the valve, there was a considerable extent of surface exposed by the long pipe in which the piston-rod was conducted through the jacket. Further, in proportion to capacity of cylinder, there was a great weight of cast iron in the cylinder, in its piston, in its valve-chest, in the pipes enclosing the piston-rod and valve-stems, and in the jacket, all of which acted either to conduct away heat, or to absorb it in raising the temperature of the metal during each steam admission the number of degrees it had lost during the remainder of the time. The specific heat of the iron thus played an important part. These causes combined were equal to the production of the enormous condensation found experimentally to exist, and furnish an explanation of the well known practical fact, that the larger the cylinder the greater is the dynamic effect produced in it by a given weight of steam, or, in other words, that one cylinder of a given area and stroke of piston will, with the same steam from the boiler used in the same manner, produce a greater dynamic effect than can be obtained from two cylinders of the same stroke and aggregate area of piston.

In the cylinders of medium size used in ordinary practice, and with steam admitted during 0·83 of the stroke of the piston and of nearly the boiler pressure, the condensation, exclusive of that which results from the production of the power, does not exceed 6 per centum, or is only about one-tenth of that in our experimental cylinder. We must not, however, overlook the fact of the steam being greatly throttled, the pressure in the cylinder at the commencement of the stroke of the piston, during the above admission, being only 59 per centum of the boiler pressure. From the commencement of the stroke to the point of cutting off, the fall of pressure was but trifling, the expansion due to throttling taking place almost entirely at the throttle-valve. So great an expansion from throttling must in itself have been attended with considerable condensation.

The injection-water, which was accurately measured, (line 36, Table No. 2,) developed directly some important facts regarding the condensation of the steam in the cylinder, and its subsequent re-evaporation,

that could not otherwise have been ascertained except by doubtful inference. Referring to line 37 of Table No. 2, and columns A, B, C, D, E, F, G, and H, it appears that the amount of injection-water which ought to have been used on the supposition that no more was required than what was needed for the condensation of the weight of steam discharged from the cylinder at the end of the stroke of its piston, averaged only 36·59 per centum of the amount actually used. Now, by referring to line 38, it appears that the amount of injection-water which ought to have been used on the supposition that all the steam which entered the cylinder exhausted from it, with the pressure shown by the indicator at the end of the stroke, averaged 90·36 per centum of the amount actually used. As the condensation due to the production of the power averaged 3·56 per centum, the resulting water of which was doubtless exhausted in suspension among the steam, there only remains $(100-\overline{90{\cdot}36\times3{\cdot}56}=)$ 6·08 per centum discrepancy for errors of observation and calculation, and for water in the state of fine spray carried out in suspension among the steam.

The following is doubtless the correct explanation of the above facts. The whole of the steam which entered the cylinder did, with the exception of a few per centum condensed and suspended among it as very fine spray, pass into the condenser in the vaporous form, where it required, of course, for its condensation the weight of injection-water experimentally used, and which, as we have seen, agreed very closely with the calculated quantity. It did not all pass to the condenser instantaneously, or nearly so, on the opening of the exhaust port at the end of the stroke of the piston, for we have seen that only 36·59 per centum of it thus passed, the remaining (90·36 — 36·59 =) 53·77 per centum passed gradually during the time the port remained open while the piston was making its return stroke. This 53·77 per centum of the steam was present in the cylinder in the form of water, at the moment when the piston reached the end of its stroke, having been condensed during the stroke and deposited upon the surfaces of the metal in the form of dew. On the opening of the exhaust port it began to re-evaporate towards the condenser under the lessened pressure, and continued until it had all passed over, carrying with it, of course, the large amount of heat which it had abstracted from the metal of the cylinder in the process of re-evaporation, and by means of which that process had been mainly effected. The heat thus abstracted from the metal had to be re-supplied during the succeeding stroke of the piston, and at the expense of the entering steam which suffered a corresponding condensation, and so on continually.

In the case of these experiments we see in a highly exaggerated degree the behavior of saturated steam in the cylinder. An enormous condensation of about 60 per centum of the steam entering the cylinder (exclusive of the condensation due to the production of the power) takes place during the stroke, and the resulting water is so completely re-evaporated during the succeeding stroke that the proper functioning of the engine is not in the slightest degree interfered with. On the contrary, the engine continues to perform its work without the least jar or shock, and without increase of temperature; and were it not for the combined evidence of the indicator diagrams, the feed-water tank measurement, and the injection-water tank measurement, the excessive cost of the power in fuel would have been an inexplicable fact, and the efficient performance by the cylinder of the functions, both of a condenser and a boiler, remained unsuspected.

If the re-evaporation were not equal to the condensation, the metal of the cylinder would receive a continual increase of temperature due to the difference. The latent heat set free by the condensation of the steam must go into the metal, and, were it not carried off by the re-evaporation as fast as imparted, would raise its temperature. If the condensation, therefore, were but a trifle greater than the re-evaporation, the metal of the cylinder would, in a short time, reach a temperature that would prevent its working. In fact, the condensation and the re-evaporation must be precisely equal; they are correlatives; the re-evapora-

tion, obviously, cannot exceed the condensation which supplies it, neither can it be less, for then the accumulated temperature in the cylinder would stop the condensation. Again, the heat imparted during condensation to the metal of the cylinder will evidently be equal to the re-conversion of the water of condensation into vapor whenever the lessened pressure lowers the boiling point of the water below the temperature of the metal on which it rests; for there has been no loss but only a transfer of heat, first, from the water in the vaporous form to the metal, and then back from the metal to the same water in the liquid form.

It has been remarked, that the proportion of the steam entering the cylinder which was condensed in it, was apparently about the same whether the steam was used with the higher or lower measure of expansion. This is contrary to general experience, for it is found in well arranged cylinders of medium size and over, that the condensation is greater when the steam is used expansively than when it is used without expansion, and greater as the measure of expansion becomes higher. In the case of our experimental engine, this fact is not apparent and could not be made so by any of the measurements taken. It can only be determined by the difference between the evaporation as measured by the tank and by the indicator. From the opening of the steam-valve up to the closing of the expansion-valve, only condensation takes place in the cylinder, but from the moment the expansion-valve closes, there occur, and simultaneously, both condensation and re-evaporation, the latter being due to the lessening pressure. Now, the indicator measures only the weight of steam actually present in the cylinder in the vaporous form at any point of the stroke of the piston, be it due to any cause whatever, as admission from the boiler, expansion proper, or re-evaporation: the difference between tank and indicator measurement, therefore, shows, not the total condensation in the cylinder, but only the difference between that condensation and the re-evaporation at the end of the stroke of the piston. The greater the condensation the more rapidly will the pressure descend during the first portion of the expansion curve, and the higher will it ascend during the last portion. This is clearly evidenced by the indicator diagrams, and must necessarily result from the fact that the quantity of re-evaporation in the cylinder is, *ceteris paribus*, governed by the reduction of the pressure, and if a greater reduction take place earlier during the stroke of the piston for a given measure of expansion, there will be a correspondingly greater re-evaporation later during the stroke. Thus in cylinders giving excessive condensation by interior surface exposure when using the steam without expansion, the disproportionally great re-evaporation occurring on the steam side of the piston, before the completion of the stroke, when using the steam expansively, might cause, as we have seen, an apparent equality in the condensation in the two cases, whereas the real condensation would prove very unequal could we find the means of ascertaining it, and as it would appear with cylinders giving but slight condensation when using the steam without expansion, in which case the condensation found when using it expansively may be properly attributed to the expansion *per se*.

2d. *Of the effect of the different pressures in the cylinder.* Confining ourselves to columns A, B, C, and D, the experiments recorded in which were made with a steam admission of 0·19 of the stroke of the piston, and observing the quantities on lines 23 and 24 of Table No. 2, we find that while the cost of the total horse power was nearly the same in all, averaging $\left(\frac{69·38128 + 66·75120 + 70·67741 + 67·56186}{4} = \right)$ 68·5930 pounds of steam, the cost of the net horse power varied from 85·7622 pounds of steam to 100·60786 pounds. The principal cause of this variation will be found on line 12 of Table No. 2, which contains the per centum that the net pressure is of the total mean pressure on the piston during its stroke, and strikingly illustrates the advantage to be derived from the use of a *high average* pressure in the cylinder, the

economic gain being sensibly in the proportion of the quantities on line 12. This is a point easily comprehended. The useless resistances of back and friction pressures being constant, and only the surplus of the total pressure over them being utilized in overcoming the resistance of the load, it is obvious that the economic effect, *ceteris paribus*, will be directly as this surplus. The difference in economy between using a mean total pressure of 13·7 pounds per square inch of piston (column D, line 9, Table No. 2), and of 22·8 pounds (column C, line 9, Table No. 2) is $\left(\frac{80{\cdot}26 - 67{\cdot}16 \times 100}{80{\cdot}26} = \right)$ 16·3 per centum of the latter.

The experiments recorded in columns E, F, G, and H, of Table No. 2, show the same fact. In these experiments the steam was admitted during 0·83 of the stroke of the piston, and the quantities on line 23 show that, while the cost of the total horse power was nearly the same, averaging $\left(\frac{82{\cdot}60790 + 79{\cdot}42902 + 82{\cdot}78948 + 79{\cdot}95424}{4} = \right)$ 81·1952 pounds of steam, the cost of the net horse power varied from 98·85420 to 117·82953 pounds. It was, in fact, sensibly in the ratio of the quantities on line 12, showing the economic gain due to this cause to be independent of the measure of expansion used for the steam, and to exist *per se*.

3d. *With regard to the gain obtained with the higher measure of expansion used for the steam.* The comparison in this respect may be made—1st. By the cost of the total horse power in the two cases, if the purpose be to ascertain the gain for the whole dynamic effect produced by the steam, irrespective of useless and useful resistances overcome. 2d. By the cost of the net horse power under the condition that the same steam pressure be used at the commencement of the stroke of the piston in both cases, if the purpose be to ascertain the respective commercial values of the two measures of expansion, discriminating for the useful resistance overcome alone. Under this condition, which is the true one for a correct practical comparison as distinguished from a theoretical or abstract one, the powers developed by the same engine will necessarily be unequal with the same load. This is the proper mode of determining the true commercial values of the two measures of expansion, because it is obvious that amount of initial pressure being purely a question of boiler, whatever pressure can be used with one measure of expansion can be used with any other measure, or without expansion, the area of the piston being properly proportioned to the load to admit of this equality of initial cylinder pressure.

And 1st. Comparing the total powers in rapport of their cost in weight of steam. We have seen that during the experiments recorded in columns A, B, C, and D, in which the steam was cut off at 0·19 of the stroke of the piston from the commencement, the total horse power cost 68·5930 pounds of steam, and that during the experiments recorded in columns E, F, G, and H, in which the steam was cut off at 0·83 of the stroke, the total horse power cost 81·1952 pounds of steam: the gain due to the higher measure of expansion was, therefore, only $\left(\frac{81{\cdot}1952 - 68{\cdot}5930 \times 100}{81{\cdot}1952} = \right)$ 15·52 per centum of the cost with the lower measure.

2d. In comparing the net powers in rapport of their cost in weight of steam under the condition of equal cylinder pressure at the commencement of the stroke of the piston in both cases, we find a wide variation from the above result, extending, in fact, to a reversal of the gain.

The experiments in columns D and F approach the nearest in proper conditions for a comparison. In both the cylinder was felted within the jacket, and the difference in initial steam pressure was only 1·2 pounds per square inch, being 23·5 pounds with the higher and 24·7 pounds with the lower measure of expansion above zero. The mean net pressure in the two cases was, respectively, 9·2 and 18·4 pounds per

square inch of piston, and the sum of the back and friction pressures was $(2{\cdot}5+2{\cdot}0=)$ 4·5 pounds per square inch of piston in both. With these proportions the cost of the net horse power with the steam cut off at 0·19 of the stroke of the piston from the commencement, was 100·60786 pounds, and with the steam cut off at 0·83 of the stroke 98·85420 pounds of steam, or $\left(\frac{100{\cdot}60786-98{\cdot}85420\times100}{100{\cdot}60786}=\right)$ 1·74 per centum in favor of the lower measure of expansion.

In this connexion it must not be forgotten that, when using the steam with the higher measure of expansion, an engine of about double size, and consequently double weight and cost, would be required to develop the same power as with the lower measure of expansion.

The gain of 15·52 per centum found for the higher measure of expansion when the total powers are compared, is larger than is found in practice under normal conditions between cutting off at 0·19 and 0·83 of the stroke of the piston. It is probably attributable to the fact, already discussed, of the proportionally greater re-evaporation with the higher measure, in the case of the experimental engine, than occurs with engines under the conditions of good practice.

4th. *Of the influence of the felting on the cylinder within the jacket.* This can be determined by a comparison of the cost of the total horse power in pounds of steam in the two cases of the felting and of its omission. With the steam cut off at 0·19 of the stroke of the piston, and with the cylinder felted within the jacket (columns B and D), the cost of the total horse power averaged $\left(\frac{66{\cdot}75120+67{\cdot}56186}{2}=\right)$ 67·15653 pounds, and with the cylinder not felted within the jacket (columns A and C) this cost averaged $\left(\frac{69{\cdot}38128+70{\cdot}67741}{2}=\right)$ 70·02934 pounds; difference in favor of felting the cylinder inside the jacket $\left(\frac{70{\cdot}02934-67{\cdot}15653\times100}{70{\cdot}02934}=\right)$ 4·10 per centum of the cost with the felting omitted.

Again, with the steam cut off at 0·83 of the stroke of the piston, and with the cylinder felted inside the jacket (column F), the cost of the total horse power was 79·42902 pounds, while this cost in the two nearest comparable experiments (columns E and G) averaged $\left(\frac{82{\cdot}60790+82{\cdot}78948}{2}=\right)$ 82·69869 pounds; difference in favor of felting the cylinder inside the jacket $\left(\frac{82{\cdot}69869-79{\cdot}42902\times100}{82{\cdot}69869}=\right)$ 3·95 per centum of the cost with the felting omitted, or sensibly the same as before.

ADHEATED STEAM USED WITH AIR IN THE JACKET. With the excessive condensation undergone by saturated steam in the experimental engine, it is evident there was ample opportunity for great improvement by the application of any means that would prevent even a moderate portion of it. Accordingly two experiments were tried (recorded in columns I and J, Table No. 2) with steam adheated in the tubular adheater by WATERMANN'S system, and used in the cylinder with air in the jacket. It will be remembered that no condensation could take place between the adheater and the cylinder, as the adheating of the steam continued up to its delivery into the cylinder valve-chest.

The two experiments were the same in all respects, except in the degree of adheating imparted. In the first (column I, Table No. 2), the boiler pressure (line 1) was 57·3 pounds per square inch above zero, the temperature of which was 289·7° Fahr., and these were sensibly the pressure and temperature in the exterior compartment of the adheater. In its interior compartment, the pressure was 34·9 pounds per square inch, sensibly the same as in the cylinder valve-chest (line 3, Table No. 2); the temperature due to this pressure as saturated steam, was 259·1° Fahr. Now its temperature as shown by the thermometer was

275° Fahr.; showing that it had been adheated (275·0—259·1=)15·9°. Under these conditions the total horse power (line 23, Table No. 2) cost 66·04878 pounds of steam. Without the adheating, using the steam with the same measure of expansion, and with the cylinder felted within the jacket, this cost was (mean of columns B and D, Table No. 2) 67·15653 pounds, making a gain for the adheater of only $\left(\frac{67{\cdot}15653 - 66{\cdot}04878 \times 100}{67{\cdot}15653} = \right)$ 1·65 per centum.

On referring to line 29 of Table No. 2, we observe that the water of condensation drawn from the adheater was 2·65 per centum of the water evaporated in the boiler. This 2·65 included the condensation due to external refrigeration in the steam-pipe connecting the boiler and the adheater, and in adheater itself. There is no certainty, however, that the condensation in the adheater did not greatly exceed this amount. It might easily have done so and the water of condensation been carried on with the steam-current in the state of fine spray suspended among the steam. When steam is enclosed in a metallic envelope, which is surrounded by a less temperature than that of the enclosed steam, the condensation is not solely at the contact of the steam and its envelope; it is likewise throughout the whole mass of the steam, every particle of which radiates heat towards the less temperature surrounding the mass. Hence it follows that the accession of 15·9° of temperature to the steam within the interior compartment of the adheater might have been purchased by a greater condensation than 2·65 per centum of the steam in the exterior compartment. The practical results seem to have been that, with a difference in the temperature of the steam in the interior and exterior compartments of the adheater considered as saturated steam of (289·7—259·1=) 30·6°, the adheating surface was able to transmit 15·9°, or 52 per centum of the difference, and that this degree of adheating operated a net gain in the cost of the power of 1·65 per centum.

In observing the other experiment recorded in column J, Table No. 2—the only difference between which and the one recorded in column I consists in its higher boiler pressure of 72·2 pounds per square inch instead of 57·3 pounds, and, consequently, a greater difference between the temperature of the steam in the two compartments of the adheater—we perceive a falling off in the economic result, the total horse power being obtained for 66·95753 pounds of steam instead of 66·04878. Comparing this result with the mean of columns B and D, as before, we find for the gain with the adheating only

$$\left(\frac{67{\cdot}15653 - 66{\cdot}95753 \times 100}{67{\cdot}15653} = \right) 0{\cdot}3 \text{ per centum.}$$

The temperature of the steam in the exterior compartment of the adheater (practically that of the boiler, line 1, Table No. 2) was 305° Fahr. The temperature of the steam in the interior compartment, considered as saturated, would be (practically that of the valve-chest, line 3, Table No. 2) 260·3° Fahr.: its actual temperature by the thermometer was 292° Fahr. It had, therefore, acquired (292—260·3=) 31·7° Fahr. of temperature. The difference between the temperature of the steam in the two compartments of the adheater, considered as saturated, was (305—260·3 =) 44·7° Fahr. The adheating surface had, therefore, transmitted 71 per centum of the difference, a much greater proportion than in the previous experiment; yet, notwithstanding this, the net gain was less. If we suppose the experimental results rigorously accurate, the only manner in which the difference can be accounted for is the greater condensation of the steam in experiment J due to its greater expansion at the throttle-valve, which was sufficient to neutralize the advantage of the higher adheating. In experiment I the expansion of the steam at the throttle-valve was $\left(\frac{57{\cdot}3}{34{\cdot}9} = \right)$ 1·64 time. In experiment J it was $\left(\frac{72{\cdot}2}{35{\cdot}6} = \right)$ 2·03 times.

It appears, then, on the whole, so great were the refrigerating influences in the cylinder, that an adheat-

ing of 31·7° possessed by the steam on entering the valve-chest, obtained by WATERMANN'S system of throttling, was inadequate to the production of any net gain in the cost of the power.

SATURATED STEAM USED WITH STEAM IN THE JACKET. The experiments recorded in columns K, L, M, and N of Table No. 2 were made with saturated steam used with steam in the cylinder-jacket. In experiments K and L the steam was delivered from the boiler first into the jacket, which served as a priming-box and received all the water—if there were any—brought over in the vesicular state by the steam, and also the water of condensation due to external refrigeration in the steam-pipe. As the capacity of the jacket was very large, comparably with that of the cylinder, and as the steam was greatly throttled in passing from the jacket to the valve-chest, the current through the jacket must have been very sluggish, giving opportunity for the steam to deposit whatever water it may have had in suspension. In the two experiments the initial steam pressure was nearly the same, but the cylinder-jacket pressure varied considerably.

In experiments M and N, the steam was delivered direct from the boiler into the cylinder valve-chest, carrying with it into the cylinder whatever water in the vesicular state it may have brought from the boiler, and also the water of condensation in the steam-pipe. In these two experiments the initial steam pressure was nearly the same, and but a little greater than in the two previous experiments. The cylinder-jacket pressures varied, but much less than in the preceding two experiments.

We have, then, to compare for total powers the economic gain due—1st. To the difference of steam pressure in the jacket. 2d. To the difference between delivering the boiler steam directly to the valve-chest, and indirectly to it through the jacket. 3d. To the use of steam in the jacket compared with the use of air in the jacket.

1st. *Of the economic gain due to the difference of steam pressure in the jacket.* In observing the cost of the total power (line 23, Table No. 2), it appears that both in columns K and M, in which the steam pressure in the jacket was greater than in columns L and N, the economy was less. In comparing K and L it appears that the latter, with 16·5 pounds per square inch less jacket pressure, was $\left(\frac{44{\cdot}97894 - 43{\cdot}72426 \times 100}{44{\cdot}97894} = \right)$ 2·79 per centum more economical. If the comparison be made for the cost of the total power (line 26, Table No. 2) exclusive of the weight of steam condensed in the jacket, &c., the economy is sensibly the same in both cases, namely, 34·01644 and 33·99416 pounds of steam per total horses power.

In comparing the cost of the total power (line 23, Table No. 2) in experiments M and N, it appears that the latter with 2·6 pounds per square inch less pressure in the cylinder-jacket was $\left(\frac{48{\cdot}41346 - 46{\cdot}97106 \times 100}{48{\cdot}41346} = \right)$ 5·04 per centum more economical. And if the comparison be made for the cost of the total power (line 26, Table No. 2), exclusive of the weight of steam condensed in the cylinder-jacket, &c., the economy is still $\left(\frac{37{\cdot}21702 - 35{\cdot}89189 \times 100}{37{\cdot}21702} = \right)$ 3·56 per centum greater.

These results agree with those in columns I and J, in which the higher exterior pressure and, consequently, adheating temperature were found to give the lowest results. Admitting the accuracy of the data, these results can only be accounted for by the greater condensation attending the greater throttling with the higher pressures. In any event they serve to show that there is no practical gain in WATERMANN'S system of adheating by throttling, attending the use of an exterior pressure much higher than the interior one.

2d. *Of the economic gain due to the difference between delivering the boiler steam directly to the valve-chest, and indirectly to it through the jacket.* To determine this gain we will compare the cost of the total horse power (line 23, Table No. 2) in columns K and L with the cost of the same power in columns M and N, taking the average of the two columns in both cases.

With the steam delivered indirectly to the valve-chest (columns K and L) the total horse power cost $\left(\frac{44{\cdot}97894 + 43{\cdot}72426}{2} = \right)$ 44·35160 pounds of steam. With it delivered directly (columns M and N) the cost was $\left(\frac{48{\cdot}41346 + 46{\cdot}97106}{2} = \right)$ 47·69226 pounds conseqently the gain effected by the indirect delivery was $\left(\frac{47{\cdot}69226 - 44{\cdot}35160 \times 100}{47{\cdot}69226} = \right)$ 7 per centum.

The reason of this difference of 7 per centum is doubtless to be found in the additional refrigerating influence caused in the cylinder by the evaporation, during the time the exhaust port is open to the condenser, of the water carried in from the condensation in the steam-pipe, and from the boiler in the vesicular state by the steam current.

On referring to the per centum of the steam evaporated in the boiler that was condensed in the cylinder jacket and steam-pipe (line 29, Table No. 2), there seems to have been no appreciable difference in the two cases; the average with the steam delivered indirectly being $\left(\frac{24{\cdot}38 + 22{\cdot}26}{2} = \right)$ 23·32 per centum; and with it delivered directly $\left(\frac{22{\cdot}64 + 23{\cdot}59}{2} = \right)$ 23·12 per centum. This equality, however, does not militate against the above difference of 7 per centum, because this 23 per centum may be the measure of all the heat the metal of the cylinder would transmit under the conditions.

Again observing the quantities on line 32 of Table No. 2, we see that the condensation in the cylinder not accounted for by the indicator averaged, with the steam delivered indirectly into the valve-chest, (columns K and L) 31·80 per centum; while with the steam delivered directly into it (columns M and N) the average was 34·83 per centum, showing that there was 3·03 per centum more condensation in the latter than in the former case.

The injection-water also demonstrates the same fact. Referring to line 38 of Table No. 2, we see that the per centum which the weight of injection-water actually used was of the weight that should have been used had all the steam which entered the cylinder passed to the condenser in the vaporous form averaged, when the steam was delivered indirectly into the valve-chest, $\left(\text{columns K and L } \frac{89{\cdot}63 + 90{\cdot}38}{2} = \right)$ 90·00; while when the steam was delivered directly into it, the average was only $\left(\text{columns M and N } \frac{89{\cdot}38 + 86{\cdot}34}{2} = \right)$ 87·86 per centum. In the former case an average of 7·70 per centum was condensed to produce the total power developed by the engine, (line 31, Table No. 2) which added to the 90·00 per centum, leaves only $(100 - \overline{90{\cdot}00 + 7{\cdot}70} =)$ 2·3 per centum unaccounted for. In the latter case the per centum of condensation due to the production of the power averaged $\left(\frac{7{\cdot}01 + 7{\cdot}32}{2} = \right)$ 7·16 which added to the foregoing 87·66 per centum, makes 95·02, leaving (100 — 95·02 =) 4·98 per centum unaccounted for or (4·98 — 2·30 =) 2·68 per centum more than before. All the quantities, therefore, go to show that the condensation in the cylinder was greater with the steam delivered directly into the valve-chest from the boiler than when it was delivered indirectly.

3d. *Of the economic gain due to the use of steam in the jacket instead of air.* To ascertain this gain we

will compare the cost of the total horse power with saturated steam used with air in the jacket as given by the experiments in columns B and D, in which the cylinder was felted within the jacket, with the experiments in columns M and N, in which the steam was delivered directly into the valve-chest, that being the usual manner in actual practice.

The average cost of the total horse power in experiments B and D was, as we have seen, 67·15653 pounds of steam, and the same power cost in experiments M and N an average of 47·69226 pounds. The gain being $\left(\frac{67·15653 - 47·69226 \times 100}{67·15653} = \right)$ 28·99 per centum of the former.

If the comparison be made for the experiments in columns K and L, in which the steam was delivered indirectly into the valve-chest, the gain will be $\left(\frac{67·15653 - 44·35160 \times 100}{67·15653} = \right)$ 33·96 per centum.

In these experiments the condensation in the steam pipe and cylinder-jacket, as determined by allowing the engine to stand for six hours after being thoroughly heated up and drawing off the water of condensation as fast as it formed, averaged 6 pounds per hour, or 10·31 per centum of the weight of steam evaporated in the boiler. The average condensation in the steam-pipe and cylinder-jacket during the experiments was, as we have seen, $\left(\frac{23·32 + 23·12}{2} = \right)$23·22 per centum, consequently, the steam condensed in the jacket by impartation of its heat to the cylinder was (23·22 — 10·31 =) 12·91 per centum of the whole weight evaporated in the boiler. We thus discover that an expenditure of steam in the jacket equal to 12·91 per centum of the amount evaporated in the boiler, effected an economy, when the steam was delivered from the boiler directly into the cylinder valve-chest, of 28·99 per centum; and when delivered indirectly into it, of 33·96 per centum; demonstrating that with the excessive condensation of 60 per centum in the cylinder when using saturated steam with air in the jacket, and with a very small cylinder of comparatively thin metal, exposing a large surface in proportion to capacity; the use of a steam-jacket, by acting preventatively of condensation in the cylinder, effects a saving under the most advantageous circumstances of 33·96 per centum, and under the ordinary circumstances of practice of 28·99 per centum, which otherwise would have been lost by the heat transferred from the metal of the cylinder to the condenser by the re-evaporation of the water of condensation during the exhaust stroke of the piston. Nevertheless, it appears the jacket alone was so far from being able to totally prevent condensation, that there still remained, as we have seen, an average of 31·80 per centum of condensation in the cylinder when the steam was delivered indirectly into its valve-chest; and of 34·83 per centum when delivered directly into it. These are in themselves large margins on which to effect still further saving.

Adheated Steam used with Steam in the Jacket. The experiments recorded in columns O to Z, both inclusive, were made with adheated steam used with steam in the cylinder-jacket. The adheating was effected by three different instruments, namely: By a *coil of pipe* in the jacket itself wound spirally around the cylinder and valve-chest: By a *tubular adheater* entirely exterior to the cylinder and jacket: And by a *steam-jacketed steam-pipe* between the boiler and cylinder valve-chest. The experiments with each instrument will be discussed under its appropriate head. In all of them, the excess of temperature for producing the adheating was obtained by Watermann's system of throttling, whereby the steam was caused to adheat itself. In the course of these experiments the steam was cut off at two different points of the stroke of the piston from the commencement, namely, at 0·19 and at 0·83. The cylinder pressure was greatly varied, and, also the relation between the steam pressure—and consequently temperature—on the outside and inside of the adheating instrument. And,

1st. Of Steam Adheated in the Coil. With this instrument were made the experiments recorded

in columns O, P, Q, and R, of which those in columns O and P were made with the steam cut-off at 0·19 of the stroke of the piston from the commencement, and those in columns Q and R with it cut off at 0·83 of the stroke. The data of these experiments enable us to determine,

1st. The absolute efficiency of the adheating system in the prevention of condensation in the cylinder.

2d. The minimum condensation under maximum conditions of adheating due to the expansion *per se* of the steam when cut off at 0·19 of the stroke of the piston.

3d. The absolute cost, and relative economical efficiency, in pounds of steam, of the total and net horse powers when cutting off at 0·19 and 0·83 of the stroke of the piston, under the condition of the maximum possible absence of condensation.

4th. The cost in weight of steam of the maximum possible prevention of condensation in the cylinder.

5th. The economic gain due to the use of adheated steam used with steam in the jacket over saturated steam used with air in the jacket, and with steam in the jacket.

6th. Of the influence of the maximum possible prevention of condensation in the cylinder upon the economic value of a high measure of expansion for the steam, relatively with the economic value of this high measure under the condition of maximum condensation in the cylinder.

1st. *Of the absolute efficiency of the adheating system in the prevention of condensation in the cylinder.* We will first examine the experiments recorded in columns Q and R made with the steam cut off at 0·83 of the stroke of the piston, and as the results, so far as relate to total powers, nearly agree, we shall take the mean.

Referring to line 32 of Table No. 2, we find that, with the exception of the weight of steam (line 31) condensed in the cylinder to produce the total power developed by the engine, there was present in the cylinder, in the vaporous form, at the end of the stroke of the piston, all the steam which had entered it, the discrepancy between the weights by tank measurement and by indicator measurement being only about one-half of one per centum. There could, consequently, have been no other condensation during the stroke of the piston than this for the production of the power which is absolutely unavoidable under any conditions, and we may therefore conclude that so far as the prevention of condensation in the cylinder is concerned, the adheating coil was perfectly efficient and left nothing to be desired. The most direct manner in which the fact of condensation in the cylinder can be determined is by a comparison of the weight of steam which entered it by tank measurement with the weight accounted for by indicator measurement, including in the latter the weight, ascertained by calculation from JOULE'S equivalent, condensed to produce the power. It can also be determined by comparing the quantities on lines 35 and 36 of Table No. 2. This comparison shows that the weight of injection-water, line 35, which would be required by calculation to condense the steam on the supposition that no more passed from the cylinder to the condenser than the number of cylinderfulls of the pressure at the end of the stroke of the piston, is sensibly the same as the weight (line 36) actually used by tank measurement; there follows, consequently, that at the end of the stroke of the piston all the steam which had entered the cylinder was still in the vaporous form, no water of condensation being present; consequently, on the opening of the exhaust port there could be no re-evaporation, and if no re-evaporation then scarcely any loss of heat by the refrigerating influence of the low back pressure vapor of the condenser.

Of the injection-water required to condense the total weight of feed-water, supposing it to have entered the condenser with the cylinder pressure at the end of the stroke of the piston (line 34), there has been accounted by the weight actually used by tank measurement (line 36) 91·26 per centum (line 38), and by the condensation in the cylinder due to the production of the power (line 31) 8·00 per centum, making a total of 99·26 per centum and leaving a discrepancy of only (100·00 — 99·26 =) 0·74 per centum.

Of the total weight of steam which entered the cylinder, there was condensed in the production of the power, as we have seen, 8 per centum (line 31, Table No. 2); but notwithstanding the fact of the cylinder and valve-chest being surrounded with steam of 35·4° Fahr., higher temperature than the steam within at the commencement of the stroke, and that the steam previous to entering the cylinder had been subjected to an adheating in the coil by nearly the same difference of temperature, yet none of the water of condensation due to the production of the power was re-evaporated. The explanation is doubtless to be found in the fact that this condensation took place intimately and equally throughout the entire mass of the steam, and the resulting water being thus disseminated among the steam in infinitesimally fine spray, was suspended in it and did not fall upon the metallic surfaces where it would have been re-evaporated. The difference between the temperature of the steam in the jacket and in the coil, considered as saturated steam, was 34·6° Fahr. The difference between the mean temperature of the steam in the cylinder during a double stroke of the piston considered as saturated steam, and of the steam in the jacket was 88·2° Fahr.

2d. *Of the minimum condensation under maximum conditions of adheating due to the expansion per se of the steam when cut off at* 0·19 *of the stroke of the piston.* In the immediately preceding section we have seen that with the steam cut off at 0·83 of the stroke of the piston all condensation in the cylinder, with the exception of that which was due to the production of the power, had been prevented by the system of adheating in conjunction with the use of the steam-jacket. We have, now, to examine whether with the steam cut off at 0·19 of the stroke of the piston and used under the same conditions of adheating and steam-jacketing, the same result was effected, and if not, then to trace the cause of the difference. The data for the determination of this problem will be found in columns O and P, Table No. 2, and as the total results vary but insensibly, we will take their mean for the discussion.

On referring to line 32, Table No. 2, and taking the mean of columns O and P, we find the difference between the weight of steam entering the cylinder, by tank measurement, and the weight accounted for by indicator measurement, including the condensation for the production of the power, to be $\left(\frac{11{\cdot}75+10{\cdot}80}{2}=\right)$ 11·28 per centum of the weight entering the cylinder, instead of 0·55 per centum (mean of columns Q and R, line 32, Table No. 2) which was the difference when the steam was cut off at 0·83 of the stroke of the piston. This greater condensation has taken place, too, notwithstanding the jacket pressure was greater than before, the cylinder initial pressure less, and the average steam pressure in the cylinder, and consequently temperature as saturated steam during a double stroke of the piston, less; all favorable conditions for *decreasing* instead of *increasing* the condensation in the cylinder. Now, as there only remained the difference in the measures of expansion employed in the two cases to produce this remarkable difference of result, we are forced to the conclusion that the greater amount of (11·28 — 0·55 =) 10·73 per centum of the steam entering the cylinder, which was condensed when the measure of expansion used was that due to cutting off at 0·19 of the stroke of the piston, than when the measure used was that due to cutting off at 0·83 of the stroke, must be attributed solely to the expansion *per se.* We are not, however, to infer that this 11·28 per centum is the whole of the condensation due to this measure of expansion *per se.* On the contrary, it is certainly too little; for it expresses only the difference between the condensation and the re-evaporation.

It may be asked how, with so great provision for adheating the steam, both in the cylinder and previous to its admission there, it was possible for such a condensation to exist. The answer is the same as in the case of the non-re-evaporation of the water of condensation due to the production of the power. The con-

densation by expansion *per se*, like that, takes place intimately throughout the entire mass of the steam, among which the resulting water exists as exceedingly fine spray, enabling the steam to hold much of it in suspension. It is only when the condensation due to this cause is so great that the steam, becoming overloaded with the spray, drops the portion which it cannot hold in suspension, that re-evaporation of it takes place from the metallic surfaces on which it is deposited.

Our final conclusion, then, is that, under the maximum conditions of adheating for the prevention of condensation in the cylinder, there is still a condensation of 11·28 per centum of all the steam entering, due to the expansion corresponding to cutting off the steam at 0·19 of the stroke of the piston. This strikingly shows how strong is the tendency of steam to condense as an effect of its own expansion *per se;* and it also proves that the water of condensation, due both to the expansion *per se* and to the production of the power, exists as infinitesimally small spray intimately diffused throughout the mass of the steam and held in suspension by it up to a certain point, when the weight of spray overcomes the suspending power of the steam and falls upon the interior metallic surfaces of the cylinder, where, of course, it will be re-evaporated on the external application of sufficient heat.

In the present case, the temperature of the steam in the jacket was 300·9° Fahr., and within the coil, considered as saturated steam, 255·9° Fahr., difference 45°. The mean temperature of the steam within the cylinder during a double stroke of the piston, considered as saturated steam, was 224° Fahr., difference between that and the temperature of the steam in the jacket 76·9° Fahr. With such excess of temperature and extent of adheating surface, and with the cylinder and valve-chest so completely immersed, it must be admitted that the conditions for adheating were a maximum, and that the application of still more external heat would probably have failed to reduce the 11·28 per centum condensation due to the expansion *per se*.

It is possible that steam might be sufficiently adheated to enable it to part with enough heat for the production of the power, and for the expansion *per se*, without falling to the temperature due to it as saturated steam for the existing pressure, in which case there could obviously be no condensation any more than there is in the use of the fixed gases, which are merely highly superheated vapors; but during these experiments and under the most favorable conditions for adheating, the steam never acquired sufficient temperature additional to what belonged to it as saturated steam to exhibit this result.

A further proof that the 11·28 per centum condensation due to the expansion *per se*, and also the 10·13 per centum (line 31, Table No. 2), due to the production of the power, passed from the cylinder to the condenser as water, without re-evaporation during the time the exhaust port remained open, is afforded by the weight of injection-water used. Comparing lines 35 and 36 of Table No. 2, we see that the weight of injection-water actually used (line 36) was sensibly what it should be (line 35), on the supposition that no more steam entered the condenser than the number of cylinderfulls of the pressure at the end of the stroke of the piston. And that of the total weight of injection-water required to condense all the steam evaporated in the boiler, supposing it to pass to the condenser in the vaporous form, with the pressure at the end of the stroke of the piston, 79·11 per centum (line 38) was actually used, leaving to pass to the condenser in the form of water (100·00—79·11=) 20·89 per centum, of which 10·13 per centum was due to the production of the power, and 11·28 per centum to the expansion of the steam *per se*, making a discrepancy of only $(20{\cdot}89 - \overline{10{\cdot}13 + 11{\cdot}28} =)$ 0·52 per centum. It is evident that the whole of this 20·89 per centum must have gone out of the cylinder into the condenser suspended among the steam as fine spray: it could never have touched any interior surface of the cylinder without being re-evaporated by the external heat in the jacket, in which event there would have been used 20·89 per centum more injection-water than the quantity experimentally found by measurement.

3d. *Of the absolute cost and relative economical efficiency in pounds of steam, of the total and net horse powers when cutting off at* 0·19 *and at* 0·83 *of the stroke of the piston, under the condition of the maximum possible absence of condensation.* Referring to line 23, Table No. 2, and taking the mean of columns O and P, we find for the cost of the *total* indicated horse power when cutting off the steam at 0·19 of the stroke of the piston 37·15161 pounds of steam, and when cutting off at 0·83 of the stroke (mean of columns Q and R) 42·66123 pounds, including in these quantities the weight of steam condensed in the cylinder-jacket and steam-pipe. Hence the higher measure of expansion was most economical by $\left(\frac{42{\cdot}66123-37{\cdot}15161\times 100}{2}=\right)$ 12·91 per centum of the lower measure. This, it will be observed, was a very small gain for so great a difference between the measures of expansion, and with adheated steam, too, used with a highly efficient steam-jacket.

Now, on referring to line 24 of the same table, and taking the mean of the same columns, we find for the cost of the *net* horse power when cutting off at 0·19 of the stroke 49 05115 pounds of steam, and when cutting off at 0·83 of the stroke, 49·92589 pounds of steam; hence the higher measure of expansion, when the comparison is made for about equal cylinder initial pressures (line 4, Table No. 2), and for commercial or practical value, is more economical than the lower by only $\left(\frac{49{\cdot}92589-49{\cdot}05115\times 100}{49{\cdot}92589}=\right)$ 1·75 per centum of the latter. This is a very striking result, and entirely opposed to common belief. It is a very accurate experimental determination that, even with the advantage of adheated steam and a much more thorough steam-jacketing than can be obtained in practice, with a cylinder giving an excessive condensation and using steam of about the average initial pressure with condensing engines, and having the minimum back and friction pressures under proper comparable conditions, the gain by the high measure of expansion over the low one amounted to only the insignificant quantity of 1·75 per centum of the latter.

Referring, now, to line 26 of Table No. 2, and taking the mean of columns O and P, we find for the cost of the *total* indicated horse powers when cutting off the steam at 0·19 of the stroke of the piston, 25·83281 pounds of steam, and when cutting off at 0·83 of the stroke (mean of columns Q and R), 34·11702 pounds of steam. These quantities are exclusive of the weight of steam condensed in the cylinder-jacket and steam-pipe. Hence it appears that, exclusive of the cost of adheating the steam and jacketing the cylinder, the economic gain by the higher measure of expansion is $\left(\frac{34{\cdot}11702-25{\cdot}83281\times 100}{34{\cdot}11702}=\right)$ 24·28 per centum of the cost with the lower one.

Making the comparison for the *net* horse powers (line 27, Table No. 2, and taking the mean of the same columns as above), we find the cost, exclusive of the weight of steam condensed in the cylinder-jacket and steam-pipe, of net horse power when cutting off the steam at 0·19 of the stroke of the piston, to be 34·10689 pounds of steam, and when cutting off at 0·83 of the stroke, 39·94349 pounds of steam. Hence it appears that, with nearly equal cylinder initial pressures, and with minimum back and friction pressures, the average pressure on the piston during the stroke with the high measure of expansion being about that of ordinary practice with condensing engines, the gain by the higher measure of expansion is only $\left(\frac{39{\cdot}94349-34{\cdot}10689\times 100}{39{\cdot}94349}=\right)$ 17·11 per centum of the cost with the lower one.

The foregoing results indicate, so far as it can be determined with an engine of the type and peculiarities of our experimental one, that, even under purely abstract conditions, that is with the steam maintained as a gas without loss of heat by either the external radiation or internal condensation of the cylinder, and that this maintenance could be effected without cost of fuel, very little economic gain would result from the

use of high over low measures of expansion, employing in all cases the same cylinder initial pressure with the average boiler, back, and friction pressures of ordinary practice.

4th. *Of the cost in weight of steam of the maximum possible prevention of condensation in the cylinder.* Taking the means of columns O and P, Table No. 1, in which experiments the steam was cut off at 0·19 of the stroke of the piston, and referring to line 15, we find the absolute condensation in the cylinder-jacket and steam-pipe to have been 13·700 pounds of steam per hour, or 30·46 per centum (line 29, Table No. 2, columns O and P) of all the steam evaporated in the boiler. The temperature of the steam in the jacket was (line 14, Table No. 2) 300·7° Fahr., and of the engine and boiler room (line 13, Table No. 2) 71° Fahr., difference 229·7° Fahr. The absolute condensation by external radiation in the cylinder-jacket and steam-pipe was 7·12 pounds of steam per hour, or 15·83 per centum of the total weight of steam evaporated in the boiler, leaving (30·46 — 15·83 =) 14·63 per centum imparted to the steam within.

Taking, in the same manner, the corresponding quantities for columns Q and R, in which the steam was cut off at 0·83 of the stroke, the absolute condensation in the cylinder-jacket and steam-pipe was 13·450 pounds of steam per hour, or 20·01 per centum of the total weight evaporated in the boiler. The temperature of the steam in the jacket was 294·4° Fahr., of the engine and boiler room 75° Fahr., difference 219·4° Fahr. The absolute condensation of steam by external radiation in the cylinder-jacket and steam-pipe was 6·85 pounds per hour, or 10·06 per centum of the total weight evaporated in the boiler, leaving (20·01 — 10·06 =) 9·95 per centum imparted to the steam within.

From these figures it appears that, as the loss by external radiation was nearly equal in both cases, and as equal condensations were effected in the jacket, the quantity of heat imparted to the steam within the cylinder must have been sensibly the same in the cases of cutting off at 0·19 and at 0·83 of the stroke of the piston. It was probably the full measure of the power of the metal of the coil and cylinder to transmit or of the steam within to absorb. But owing to the greater quantity of steam used in equal times when cutting off at 0·83 of the stroke than when cutting off at 0·19, it appears a larger per centum of the latter than of the former.

It will be observed from the above that, with the low measure of expansion, an adheating equal to 9·95 per centum of the heat employed to evaporate the steam in the boiler, prevented all condensation in the cylinder other than that due to the production of the power, while, with the high measure of expansion, an adheating equal to 14·63 per centum of the heat employed to evaporate all the steam in the boiler was insufficient to prevent condensation in the cylinder, which remained 11·28 per centum in addition to that due to the production of the power. Nothing could more strikingly exhibit the strong tendency of steam to condense as an effect of its own expansion—apart from overcoming external resistance.

5th. *Of the economic gain due to the use of adheated steam used with steam in the jacket over saturated steam used with air in the jacket, and with steam in the jacket.* We have seen that the cost of the total horse power with saturated steam used with air in the jacket, and cutting off at 0·19 of the stroke of the piston, was 68·59300 pounds of steam per hour, and that with the same point of cutting off, but using adheated steam with steam in the cylinder-jacket, the cost of the same power was 37·15161 pounds of steam per hour, inclusive of the condensation in the cylinder-jacket and steam-pipe, the latter method of using steam is consequently the most economical by $\left(\frac{68{\cdot}59300 - 37{\cdot}15161 \times 100}{68{\cdot}59300} = \right)$ 45·84 per centum of the former. Whence it appears that an expenditure in adheating of 14·63 per centum of the heat in the steam evaporated in the boiler effected a saving of 45·84 per centum in the cost of the power.

When the steam was cut off at 0·83 of the stroke of the piston, the cost of the total horse power, using

saturated steam in the cylinder with air in the jacket, was 81·19520 pounds of steam per hour, while with the adheated steam used with steam in the jacket this cost was only 42·66123 pounds of steam per hour, the latter is, consequently, most economical by $\left(\frac{81 \cdot 19520 - 42 \cdot 66123 \times 100}{81 \cdot 19520} = \right)$ 47·46 per centum, or slightly more than before. Whence it appears that an expenditure in adheating of 9·95 per centum of the heat in the steam evaporated in the boiler effected a saving of 47·46 per centum of the cost of the power.

The cost of the total horse power, when saturated steam is used in the cylinder, and cut off at 0·19 of the stroke of the piston, with steam in the jacket, the steam from the boiler being delivered indirectly to the valve-chest through the jacket, as in the case of the adheated steam used with steam in the jacket (columns K and L, Table No. 2), was 44·35160 pounds of steam per hour; consequently the adheated steam, cutting off at the same point and used with steam in the jacket, was most economical by

$$\left(\frac{44 \cdot 35160 - 37 \cdot 15161 \times 100}{44 \cdot 35160} = \right) \text{16·23 per centum of the former.}$$

The condensation due to external radiation from the cylinder-jacket and steam-pipe during experiments K and L, was 6·25 pounds of steam per hour, or 11·40 per centum of the steam evaporated in the boiler, leaving for the heat imparted to the steam within the cylinder (23·32—11·40 =) 11·92 per centum. In the corresponding experiments (columns O and P, Table No. 2), this condensation was, as we have seen, 14·63 per centum, whence it appears that the heat imparted by the coil to the steam within the cylinder, was, in addition to that imparted by the cylinder surface (14·63—11·92 =) 2·71 per centum of all that was expended in evaporation in the boiler.

6th. *Of the influence of the maximum possible prevention of condensation in the cylinder upon the economic value of a high measure of expansion for the steam, relatively with the economic value of this high measure under the condition of maximum condensation in the cylinder.* When saturated steam was used in the cylinder with air in the jacket, the condensation in the cylinder, exclusive of that due to the production of the power being 60 per centum of all the steam entering the cylinder, the difference in the economic effect, compared for cost of total horse powers, when cut off at 0·19 and at 0·83 of the stroke of the piston, was 15·52 per centum of the cost, with the latter point of cutting off in favor of the former.

Now, when adheated steam was used in the cylinder with steam in the jacket, the condensation in the cylinder, exclusive of that due to the production of the power, being 11·28 per centum of all the steam entering the cylinder when the steam was cut off at 0·19 of the stroke of the piston, and 0·55 per centum when it was cut off at 0·83 of the stroke, the difference in the economic effect, compared for cost of total horse powers, *exclusive of the steam condensed in the cylinder-jacket and steam-pipe*, was 24·28 per centum of the cost when cutting off at 0·83 in favor of the cost when cutting off at 0·19 of the stroke. It thus appears that adheating the steam, using it with steam in the cylinder-jacket, and with the minimum condensation in the cylinder, produced a gain of (24·28—15·52 =) 8·76 per centum more relatively for the high measure of expansion, than when the saturated steam was used with air in the jacket and the maximum condensation in the cylinder.

As, however, this adheating cannot be obtained for nothing, we must, for the commercial or practical value of its effect on the use of steam with a high measure of expansion relatively over a low one, make the comparison for total horse powers, *inclusive of the condensation in the cylinder-jacket and steam-pipe.* Under this condition the gain by the higher measure of expansion was only 12·91 per centum of the cost with the lower measure, which is a reversal of the previous determination, and shows that adheating the steam and using it with steam in the jacket lessened the relative difference in the economy between the two measures of expansion (15·52—12·91 =) 2·61 per centum.

2d. Of Steam Adheated in the Tubular Adheater. Columns S, T, U, V, and W, contain the results of the experiments made with steam adheated in the tubular condenser, and used with steam in the cylinder-jacket. These experiments were varied by employing the two measures of expansion practicable with the valve-gear, namely, those due to cutting off at 0·19 and at 0·83 of the stroke of the piston. They were also varied by the use of different average pressures on the piston (line 9, Table No. 2), and by different relations between the boiler and cylinder initial pressures (lines 1 and 4, Table No. 2).

If we observe the cost of the total horse power, both inclusive and exclusive of the condensation in the cylinder-jacket, tubular adheater, and steam-pipe, (lines 23 and 26, Table No. 2) it appears that no decided effect was produced on the economy by the variations either in the average pressure on the piston, or in the relation between the boiler and cylinder initial pressures.

If any benefit did really attend the employment of a lower average pressure on the piston, which, as it lessened the difference between the temperatures on the steam side of the piston and on the back pressure side, it is rational to suppose was the case, it was lost by the attending greater condensation in the cylinder-jacket and steam-pipe (line 29, Table No. 2) proportionally to the reduced power developed by the engine.

The same remark applies to the benefit that might have resulted from greater difference between the pressure in the cylinder-jacket and the cylinder initial pressure, from which, too, there is a further reduction to be made for the loss by the greater throttling.

There then remains to compare the cost of the total and the net powers, both inclusive and exclusive of the condensation in the cylinder-jacket, tubular adheater and steam-pipe, (lines 23 and 26, and 24 and 27, Table No. 2) in the two cases of cutting off the steam at 0·19 and at 0·83 of the stroke of the piston. In making these comparisons we shall take, for the total powers, the mean of the results in columns S, T, and U, for the cost with the higher measure of expansion, and in columns V and W, for the cost with the lower; and for the net powers we shall take for the cost with the higher measure of expansion the results in column U, and for the cost with the lower measure the mean of the results in columns V and W. The reason for this selection with the net powers is that the mean of the cylinder initial pressures (line 4, Table No. 2) in columns V and W is almost exactly the same as the cylinder initial pressure in column U; and this equality of cylinder initial pressure, together with equality of back and piston pressures, are essential to a just comparison between the economic effects due to the different measures of expansion.

And, first, the cost of the total horse power, inclusive of the condensation in the cylinder-jacket, tubular adheater, and steam-pipe, (line 23, Table No. 2,) when cutting off the steam at 0·19 of the stroke of the piston, is $\left(\text{mean of columns S, T, and U, } \frac{40{\cdot}59646+41{\cdot}13162+40{\cdot}87525}{3}=\right)$ 40·86778 pounds of steam per hour. And when cutting off at 0·83 of the stroke the cost is $\left(\text{mean of columns V and W, } \frac{46{\cdot}77729+46{\cdot}81600}{2}=\right)$ 46·79664 pounds, whence it appears that the higher measure of expansion was most economical by $\left(\frac{46{\cdot}79664-40{\cdot}86778\times 100}{46{\cdot}79664}=\right)$ 12·67 per centum of the lower.

Again, the cost of the total horse power, exclusive of the condensation in the cylinder-jacket, tubular adheater, and steam-pipe, (line 26, Table No. 2,) when cutting off the steam at 0·19 of the stroke of the piston, is $\left(\text{mean of columns S, T, and U, } \frac{29{\cdot}24427+28{\cdot}88162+26{\cdot}48195}{3}=\right)$ 28·20261 pounds of steam

per hour. And when cutting off at 0·83 of the stroke the cost is $\left(\text{mean of columns V and W, } \frac{37·10752+37·63220}{2}=\right)$ 37·36986 pounds. Whence it appears that the higher measure of expansion was most economical by $\left(\frac{37·36986-28·20261\times100}{37·36986}=\right)$ 24·53 per centum of the lower.

It thus appears that the result of the adheating in conjunction with the steam-jacketing, exclusive of the cost of effecting it, increased the economy due to the higher measure of expansion (24·53—12·67 =) 11·86 per centum.

In comparing the economy of the net powers with the two measures of expansion, the cylinder initial pressures being equal, we have for the cost of the net horse power, inclusive of the condensation in the cylinder-jacket, tubular adheater, and steam-pipe, when cutting off at 0·19 of the stroke of the piston, (column U, line 24, Table No. 2,) 66·42218 pounds of steam per hour, and when cutting off at 0·83 of the stroke $\left(\text{mean of columns V and W, line 24, Table No. 2, } \frac{61·70634+59·20804}{2}=\right)$ 60·45719 pounds per hour, whence it appears that the higher measure of expansion is the least economical by

$\left(\frac{66·42218-60·45719\times100}{60·45719}=\right)$ 9·87 per centum of the cost with the lower measure.

Again, making the above comparison, but exclusive of the condensation in the cylinder-jacket, tubular adheater, and steam-pipe, we have for the cost of the net horse power, when cutting off at 0·19 of the stroke of the piston, (column U, line 27, Table No. 2,) 43·03309 pounds of steam per hour, and when cutting off at 0·83 of the stroke, $\left(\text{mean of columns V and W, line 27, Table No. 2, } \frac{48·95045+47·59332}{2}=\right)$ 48·27189 pounds per hour, whence it appears that the higher measure of expansion is most economical by

$\left(\frac{48·27189-43·03309\times100}{48·27189}=\right)$ 10·85 per centum of the cost with the lower measure.

From the above it appears that the adheating, exclusive of its cost, effected (9·87 + 10·85 =) 20·72 greater increase in the economy with the higher measure of expansion than in that with the lower measure.

There remains to compare the efficiency of the tubular adheater with that of the coil as an adheating instrument. We shall make this comparison for the cost of the total powers with the two measures of expansion, and both inclusive and exclusive of the condensation in the cylinder-jacket, tubular adheater, and steam-pipe.

First, when cutting off the steam at 0·19 of the stroke of the piston, and inclusive of the condensation in the cylinder-jacket, &c., the cost of the total horse power with the coil was, as we have seen, 37·15161 pounds of steam per hour, and with the tubular adheater 40·86778 pounds, showing the latter to be the least economical by $\left(\frac{40·86778-37·15161\times100}{37·15161}=\right)$ 10·03 per centum of the former.

Again, when cutting off at 0·83 of the stroke of the piston, and inclusive of the condensation in the cylinder-jacket, &c., the cost of the total horse power with the coil was 42·66123 pounds of steam per hour, and with the tubular adheater 46·79664 pounds, showing the latter to be the least economical by

$\left(\frac{46·79664-42·66123\times100}{42·66123}=\right)$ 9·70 per centum of the former.

Second, when cutting off the steam at 0·19 of the stroke of the piston, and exclusive of the condensation

in the cylinder-jacket, &c., the cost of the total horse power with the coil was, as we have seen, 25·83281 pounds of steam per hour, and with the tubular adheater 28·20261 pounds, showing the latter to be the least economical by $\left(\frac{28{\cdot}20261-25{\cdot}83281\times 100}{25{\cdot}83281}=\right)$ 9·18 per centum of the former.

Again, when cutting off at 0·83 of the stroke of the piston, and exclusive of the condensation in the cylinder-jacket, &c., the cost of the total horse power with the coil was 34·11702 pounds of steam per hour, and with the tubular adheater 37·36986 pounds, showing the latter to be the least economical by

$$\left(\frac{37{\cdot}36986-34{\cdot}11702\times 100}{34{\cdot}11702}=\right) 9{\cdot}53 \text{ per centum of the former.}$$

From the preceding it appears that the relative economy was not practically affected either by the measure of expansion with which the steam was used, or whether the comparison be made inclusively or exclusively of the condensation in the cylinder-jacket, &c., and that the tubular adheater was less economically efficient than the coil by $\left(\frac{10{\cdot}03+9{\cdot}70+9{\cdot}18+9{\cdot}53}{4}=\right)$ 9·61 per centum of the latter.

In examining the causes of the difference in the economic efficiency of the two adheating instruments, we will first ascertain the proportion of the total steam evaporated in the boiler that was condensed in the cylinder-jacket, tubular adheater, and steam-pipe, making the inquiry separately for the two measures of expansion used. And

First, when cutting off the steam at 0·19 of the stroke of the piston, we find the condensation in the jacket, &c., with the coil, (mean of columns O and P, line 29, Table No. 2,) to have been 30·46 per centum, and with the tubular adheater (mean of columns S, T, and U,) 30·99 per centum, or sensibly the same.

Second, when cutting off at 0·83 of the stroke the condensation in the jacket, &c., with the coil, (mean of columns Q and R, line 29, Table No. 2,) was 20·01 per centum, and with the tubular adheater (mean of columns V and W) 20·15 per centum, or sensibly the same.

Hence it appears that, both with the coil and the tubular adheater, the same proportion of the total steam evaporated in the boiler was expended on external refrigeration and in adheating. But we have seen the economic result of this equal expenditure differed 9·61 per centum. Why should this be so?

If we refer to columns S, T, U, V, and W, and lines 35 and 36 of Table No. 2, the first of which lines contains the pounds of injection-water required to condense the steam on the supposition that no more steam passed from the cylinder to the condenser in the vaporous form than the number of cylinderfulls of the pressure at the end of the stroke of the piston (line 6, Table No. 2), and the second of which contains the pounds of injection-water actually used, we shall find the quantities sensibly the same, showing that when the piston had reached the end of its stroke all the steam which had entered the cylinder passed at once, either in the vaporous or liquid form, to the condenser on the opening of the exhaust passage. In other words, there was no water of condensation in the cylinder remaining to be re-evaporated during the return stroke under the less pressure of the condenser. This was also the case with the coil: why, then, should not the economic effects be the same?

Referring, now, to line 32 of Table No. 2, and taking the mean of columns S, T, and U, which contain the results when cutting off the steam at 0·19 of the stroke of the piston, we find the condensation with the tubular adheater to have been in the cylinder, exclusive of that required for the production of the power, (line 31, Table No. 2,) 22·30 per centum of all the steam which entered it. With the coil this condensation was only 11·28 per centum, difference (22·30—11·28=) 11·02 per centum against the tubular ad-

heater. We have here the cause of the difference of 9·61 per centum in the economic results produced. It was owing to the greater condensation of 11·02 per centum in the cylinder when the tubular adheater was used than when the coil was used. And this is accompanied by the other remarkable fact that the proportion of the total evaporation in the boiler condensed in the cylinder, &c., was the same in both cases, and that in both cases there remained no water of condensation in the cylinder at the end of the stroke of the piston to undergo re-evaporation during the time the exhaust passage remained open; consequently, this 22·30 per centum of all the steam which entered the cylinder, in addition to and with the 9·40 per centum (line 31) condensation to produce the power, must have passed out, when the exhaust passage opened, as water of condensation in the form of minute spray drops held in suspension among the steam, and that none of this water had been deposited on the interior surfaces of the cylinder. But why, under the above equality of conditions, should this inequality of condensation take place in the cylinder? This question is difficult to answer, but the following comparison of the surfaces and capacities of the two adheating instruments may not be useless in this connexion.

The coil contained 9·4224 square feet of interior wrought iron surface. The tubes of the tubular adheater contained 30·9961 square feet of interior brass surface, or 3·29 times as much adheating surface as the coil. The capacity of the coil, from throttle-valve to valve-chest, was 0·29448 cubic feet. The capacity of the interior of the tubes and spaces in connexion with them, from throttle-valve to valve-chest was 0·47474 cubic feet, or 1·61 times as much as the coil. The capacity of the cylinder, from steam-valve to end of stroke of piston was 0·13272 cubic feet.

It would seem that the numerous (90) straight, short (24⅝ inches long) tubes with small diameter (1½ inch internal diameter) of the tubular adheater were radically inferior as adheating surface to the long (24 feet) single, helical tube of the coil with greater diameter, (1½ inch internal.)

That there was no leakage past the cylinder piston and valve was clearly proven by direct trial, allowing the steam to stand on them for several hours with the engine at rest. It is also proven by the equality between the quantities on lines 35 and 36 of Table No. 2. This equality could not have existed had the 22·30 per centum, or any part of it, leaked to the condenser; for the injection water would then have been experimentally found to be greater than it was to a proportional degree.

Of the total amount of steam that was generated in the boiler, we have seen that 30·99 per centum (mean of columns S, T and U, line 29, table No. 2) was condensed in the cylinder-jacket, tubular adheater, and steam-pipe, leaving (100·00—30·99=) 69·01 per centum as the proportion which entered the cylinder. Of this amount 9·40 per centum (mean of columns S, T, and U, line 31, Table No. 2) was condensed to produce the power developed by the engine, 66·67 per centum (line 37, Table No. 2) was exhausted in the vaporous form into the condenser at the end of the stroke of the piston, and 22·30 per centum in addition to and with the preceding 9·40 per centum, passed to the condenser at the opening of the exhaust passage as water of condensation in the form of fine spray suspended among the steam, leaving unaccounted for 1·63 per centum due to all errors of measurement and calculation.

Again, for the case of cutting off the steam at 0·83 of the stroke of the piston, and referring to the quantities on line 32 of Table No. 2, taking the mean of columns V and W, we find the condensation in the cylinder, exclusive of that due to the production of the power, to be 12·75 per centum of all the steam that entered it. Now, under the same conditions with the coil, this condensation was only 0·55 per centum, (mean of columns Q and R, line 32, Table No. 2) difference 12·20 per centum, or nearly the same as between the two adheating instruments when the steam was cut off at 0 83 of the stroke.

The greater condensation attending the expansion of the steam *per se*, is evidenced from these results;

for the condensation in the cylinder, exclusive of that due to the production of the power, which, with the higher measure of expansion we saw amounting to 22·30 per centum, reaches, with the lower measure of expansion to only 12·75 per centum, notwithstanding all the prevention affected by the adheating and steam-jacketing. Despite these favorable conditions for the more highly expanded steam, its condensation remained 9·55 per centum greater than that of the steam used with the lower measure of expansion.

The temperature of the adheated steam as it left the tubular adheater was obtained during the experiments recorded in columns S, T, U, and V, (line 18, Table No. 2) and averaged $\left(\frac{268+270+242+244}{4}=\right)$ 256° Fahr. The temperature of steam of the pressure of that in the valve-chest (line 3, Table No. 2) during the same experiments, considered as saturated, averaged $\left(\frac{254{\cdot}4+263{\cdot}2+231{\cdot}3+228{\cdot}0}{4}=\right)$ 244·25° Fahr.; whence it appears that the tubular adheated had increased the normal temperature of the steam entering the cylinder 11·75° Fahr.

The temperature of the adheating steam in the outer compartment of the tubular adheater (sensibly the same as that in the cylinder-jacket, (line 14, Table No. 2) averaged) $\left(\frac{272{\cdot}2+277{\cdot}2+260{\cdot}1+253{\cdot}3}{4}=\right)$ 265·7° Fahr.; whence it appears that the adheated steam in the interior compartment of the tubular adheater had a temperature which was a mean between that due to its pressure as saturated steam, and the temperature of the adheating steam in the outer compartment of the tubular adheater, namely,

$\left(\frac{265{\cdot}70+244{\cdot}25}{2}=\right)$ 255° Fahr. It was by thermometer, as seen above, 256° Fahr.

As the steam from the boiler was delivered into the outer compartment of the tubular adheater during all the experiments made with it, no water primed over, had there been any, and no water of condensation due to external refrigeration in the steam-pipe, &c., could have entered the cylinder.

3d. Of Steam Adheated in the Steam-jacketed Steam-pipe. A trial was made, finally, of the effect to be obtained from the conversion of the steam-pipe connecting the boiler and cylinder valve-chest into an adheater, by enclosing it in another pipe, placing the throttle-valve at the boiler end, and keeping the annular space between the pipes filled with steam of the boiler pressure. The exterior pipe was of course well felted. A priming-box was interposed between the boiler and the inner pipe to intercept any water primed over, and the throttle-valve was placed in the mouth of the pipe adjacent to the priming-box. The total length of steam pipe thus used was 20 feet, its interior diameter was $1\frac{9}{16}$ inch, and its exterior diameter was $1\frac{13}{16}$ inch. It had but one turn, which was a right-angled elbow. Its enveloping pipe or jacket extended from the boiler to the cylinder-jacket, with both of which it communicated, and was $2\frac{1}{2}$ inches in interior diameter. With this adheating instrument the two experiments recorded in columns X and Y were made; the first with the steam cut off at 0·19 of the stroke of the piston, the second with it cut off at 0·83 of the stroke. In both cases steam was used in the cylinder-jacket.

By comparing the cost of the total powers, inclusive of the condensation in the cylinder and steam-pipe jackets (line 23, Table No. 2) it appears that the higher measure of expansion was the most economical by $\left(\frac{47{\cdot}14389-39{\cdot}97301\times100}{47{\cdot}14389}=\right)$ 15·21 per centum of the lower measure.

Again, making the same comparison, but exclusive of the condensation in the cylinder and steam-pipe jackets, (line 26, Table No. 2) it appears that the higher measure of expansion was the most economical by $\left(\frac{38{\cdot}75460-28{\cdot}08513\times100}{38{\cdot}75460}=\right)$ 27·53 per centum of the lower measure.

In comparing the cost of the total horse power, inclusive of the condensation in the cylinder and steam-pipe jackets, when cutting off at 0·19 of the stroke of the piston, with the cost of the same when the coil was employed, it appears that the steam-jacketed steam pipe was less economical by

$\left(\frac{39{\cdot}97301 - 37{\cdot}15161 \times 100}{39{\cdot}97301} = \right) 7{\cdot}06$ per centum of the cost with the coil.

Again, making the same comparison, but exclusive of the condensation in the cylinder and steam-pipe jackets, it appears that the steam-jacketed steam-pipe was less economical by

$\left(\frac{47{\cdot}14389 - 42{\cdot}66123 \times 100}{47{\cdot}14389} = \right) 9{\cdot}51$ per centum of the cost with the coil.

On referring to lines 35 and 36 of Table No. 2, it will be seen from the sensible equality of the weight of injection-water actually used, and the weight which ought to have been used on the supposition that no more steam entered the condenser in the vaporous form than the number of cylinderfulls of the pressure at the end of the stroke of the piston, that there was no re-evaporation during the return or exhaust stroke, consequently, the whole of the condensation which took place in the cylinder, amounting, when cutting off at 0·19 of the stroke of the piston to (sum of quantities on lines 31 and 32, column X, Table No. 2, namely, $9{\cdot}33 + 21{\cdot}91 =$) 31·24 per centum of all the steam which had entered the cylinder; and when cutting off at 0·83 of the stroke to ($6{\cdot}89 + 13{\cdot}51 =$) 20·40 per centum of the same; must have been suspended among the steam as fine spray and passed with it to the condenser on the opening of the exhaust passage.

When cutting off at 0·19 of the stroke of the piston it appears that, of all the steam which had entered the cylinder, 9·33 per centum was condensed in the production of the power, 21·91 per centum was condensed by other causes, and 65·61 per centum (line 38, Table No. 2) was discharged at the end of the stroke of the piston by actual measurement of injection-water, leaving a discrepancy of only

$$(100 - \overline{9{\cdot}33 + 21{\cdot}91 + 65{\cdot}61} =)\ 3{\cdot}14 \text{ per centum.}$$

When cutting off at 0·83 of the stroke of the piston, it appears that, of all the steam which had entered the cylinder, 6·89 per centum (line 31, Table No. 2, column Y) was condensed to produce the power and 13·51 per centum (line 32, same column and Table) was condensed by other causes, making a total condensation in the cylinder of ($6{\cdot}89 + 13{\cdot}51 =$) 20·40 per centum. Now, as we have just seen, the same condensation when cutting off at 0·19 of the stroke to have been 31·24 per centum—the difference of ($31{\cdot}24 - 20{\cdot}40 =$) 10·84 must have been due to the difference in the condensation by expansion *per se* with the two different measures employed, notwithstanding the adheating in the steam-pipe and cylinder caused by the steam-jackets, and that there was no refrigeration in either case by re-evaporation during the exhaust stroke of the piston. By actual measurement of injection-water, there was discharged into the condenser at the end of the stroke of the piston (line 37, column Y, Table No. 2) 79·61 per centum of all the steam which had entered the cylinder, leaving no discrepancy at all, as $100 - \overline{20{\cdot}40 + 79{\cdot}61} = 0{\cdot}01$.

In experimenting with the two measures of expansion, the same mean total pressure (line 9, Table No. 2) was used in the cylinder with the same speed of piston, the difference in the cylinder initial pressures being sufficiently great to produce this equality of result with the same load. Under these conditions the difference in the condensation in the steam jackets in the cylinder and steam-pipe was remarkable, being, of the total weight of water pumped into the boiler, 29·74 per centum when the steam was cut off at 0·19 of the stroke of the piston, and 17·79 per centum when cut off at 0·83 of the stroke. Part of this difference was, of course, due to the greater external refrigeration when cutting off at 0 19, on account of

the higher pressure of steam in the jackets, and the remainder was due to the greater internal refrigeration caused by the greater expansion *per se.*

GENERAL DEDUCTION FROM THE EXPERIMENTS. From the experiments hereinbefore detailed and discussed, the following general deductions seem to be warranted, namely:

1st. That the cost of power in steam engines is greatly affected by the absolute size of the cylinder and its connexions, and by the thickness of the metal of which it is constructed, as approximative causes. The cost increasing as the size decreases, and as the thickness of metal increases; not, however, in a direct ratio but probably in the ratio of some power of the proportionality of the interior surface, and of the thickness of the metal, to the capacity. It does not appear that, *ceteris paribus,* the cost is modified by the greater or less density of the steam used.

From the size of the experimental engine and the disproportionately great thickness of its metal, and from the nature and weight of its appendages, the cost of the power derived through it may be regarded as a maximum; and all the effects attending its working must, from the above causes, be considered as highly exaggerated in *degree,* though the same in *kind* as those found with the largest cylinders operating under normal conditions; they will, therefore, mark the laws with greater certainty, but will fail to determine numerical values applicable to larger engines under the conditions of ordinary manufacture and working. These considerations must be carefully borne in mind when studying the results of these experiments. A disregard of them has led to many false hypotheses, blasted hopes, and fruitless expenditures of talent, time, labor and money.

2d. That the ultimate cause of the difference in the cost of power with large and small cylinders is, with the exception of difference in the proportion of friction to power, to be found in the fact of the alternating action of the cylinder as condenser and as boiler, which produces a periodic heating, and refrigeration of the interior surfaces by the re-evaporation from them—mainly at the expense of the heat previously imparted to them by the entering steam—of the water deposited upon them as dew by the condensation of the entering steam which thus imparts sufficient heat to again equalize their temperature with its own; and so on continually. The smaller cylinder exposing, proportionally to capacity, much more surface for this action, must have a greater loss, proportionally, than the larger cylinder.

When steam of less density is used in the same cylinder, although the proportion of surface to power increases, yet as the difference between the temperatures on the two sides of the piston diminishes, the loss due to the first is neutralized by the gain caused by the last.

3d. The loss of heat by the metal of the cylinder during a double stroke of the piston, apart from that due to the above re-evaporation, is caused by external radiation, and by the difference between the temperature of the metal on the exhaust side of the piston and of the back pressure vapor resting on it, which back pressure absorbs as a gas some of the heat and transfers it to the condenser, for a cylinderfull of back pressure vapor with all the heat it has extracted from the metal, is pushed into it at each stroke of the piston.

4th. As a corollary to 3d., and also directly proven by the experiments, the cost of power is affected by the more or less thorough clothing of the cylinder.

5th. That the cost of power is greatly affected by the absolute average cylinder pressure employed; the higher pressure having the greater economy.

With equal back and friction pressures, and different average total pressures, the economy appears to

be, with the same measure of expansion, sensibly in the direct ratio of the remainders of the average total pressures after deduction of the back and friction pressures.

6th. That the cost of power is lessened by even slightly adheating the steam previous to its admission to the cylinder.

7th. That the cost of power is lessened by enveloping the cylinder in a steam-jacket. The gain due to this appendage will, *ceteris paribus*, be greater with smaller than with larger engines. In the case of our experimental engine, with which this gain must be considered a maximum, it was 29 per centum, using the steam cut off at 0·19 of the stroke of the piston, and having a condensation in the cylinder of 59 per centum to act on preventatively.

8th. That the cost of power is lessened by preventing the introduction of water into the cylinder from outside. Hence a gain is derived by employing a priming-box between the cylinder valve-chest and the steam-pipe, in which box the steam is separated from the water of condensation of the pipe and the water brought over from the boiler, if there be any.

9th. That the maximum economy in the cost of power is obtained by both adheating the steam before its introduction into the cylinder, and by enveloping the cylinder in a steam-jacket. The combination is essential.

10th. That in effecting the adheating, according to WATERMANN'S system, by differences produced in the temperature of the steam after it has left the boiler, by throttling it from one side of a surface to the other, there appears no advantage from a great difference in the two temperatures, the condensation due to the expansion *per se* by the throttling, probably neutralizing the advantage of the greater difference between the temperatures. *Time* of contact seems the essential element for maximum economy.

11th. That of the three adheating instruments experimented with, namely, the coil, the tubular adheater, and the steam-jacketed steam-pipe, the first was more economical than the last two, which were sensibly equal in point of economy.

NOTE. The numerical values in pounds of steam per hour of the total indicated horse power, with the experimental engine, under its different methods of using the steam, will be found in the following table, in which the cost is given, both inclusive and exclusive of the condensation in the steam-jackets and tubular adheater, and for the different measures of expansion employed.

| | | COST OF THE TOTAL INDICATED HORSE POWER. | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | Inclusive of Condensation in Steam-Jackets and Tubular Adheater. | | | | Exclusive of Condensation in Steam-Jackets and Tubular Adheater. | | | |
| | | Steam cut off at 0·19. | | Steam cut off at 0·83. | | Steam cut off at 0·19. | | Steam cut off at 0·83. | |
| | | Pounds of Steam per hour. | Proportional. | Pounds of Steam per hour. | Proportional. | Pounds of Steam per hour. | Proportional. | Pounds of Steam per hour. | Proportional. |
| Saturated Steam with Air in Cylinder-Jacket, | | 67·15653 | 1·000 | 79·42902 | 1·000 | 67·15653 | 1·000 | 79·42902 | 1·000 |
| Adheated Steam with Air in Cylinder-Jacket, | | 66·50316 | 0·990 | . . | . . | 64·88411 | 0·966 | . . | . . |
| Saturated Steam with Steam in Cylinder-Jacket, | | 44·35160 | 0·664 | . . | . . | 34·00530 | 0·506 | . . | . . |
| Steam Adheated in Coil, | Cylinder | 37·15161 | 0·553 | 42·66123 | 0·537 | 25·83281 | 0·385 | 34·11702 | 0·429 |
| Steam Adheated in Tubular Adheater, | Steam- | 40·86778 | 0·608 | 46·79664 | 0·589 | 28·20261 | 0·420 | 37·86986 | 0·470 |
| Steam Adheated in Steam-Pipe, | Jacketed. | 39·97301 | 0·595 | 47·14389 | 0·594 | 28·08513 | 0·418 | 38·75460 | 0·488 |

12th. That when the cost of the power is reckoned inclusively of the condensation in the steam-jackets and tubular adheater, the economical gain given by the adheating and steam-jacketing is unaffected by the measure of expansion used.

13th. That when the cost of the power is reckoned exclusively of the condensation in the steam-jackets and tubular adheater, the economical gain given by the adheating and steam-jacketing is greater with the higher than with the lower measure of expansion.

14th. That in the case of the maximum economy, namely, that of the coil in combination with the cylinder steam-jacket and cutting-off at 0·19 of the stroke of the piston, it appears the cost of the total indicated horse power, reckoned exclusively of the condensation in the cylinder-jacket and steam-pipe, was 25·83281 pounds of steam per hour, with a remaining condensation in the cylinder at the end of the stroke of the piston of 11·28 per centum, exclusive of that due to the production of the power, of all the steam that had entered it. If, now, the adheating had been sufficient to entirely prevent condensation, then the cost of the total indicated horse power, exclusive of all condensations whatever, except the unavoidable one due to the production of the power, and consequently with the steam maintained in the gaseous state from the commencement to the end of the stroke of the piston at no expense, would have been

$$(25{\cdot}83281 \times \overline{100{\cdot}00 - 11{\cdot}28} =)\,22{\cdot}91887 \text{ pounds of steam per hour.}$$

The net indicated horse power would, of course, cost more. To estimate it, suppose the average total pressure on the piston to be 25 pounds per square inch, which is about the highest found in practice with condensing engines, and that the sum of the back and friction pressures is 5 pounds per square inch, then the net pressure would be 80 per centum of the total pressure, and the cost of the net indicated horse power would become $\left(\frac{22{\cdot}91887}{0{\cdot}80}=\right)$ 28·64859 pounds of steam per hour. Now, supposing the pound of coal to evaporate 9·5 pounds of water from the temperature of 100° Fahr., which requires it to be of the best quality and burned in a boiler of the highest evaporative efficiency, then the cost of the net indicated horse power will be $\left(\frac{28{\cdot}64859}{9{\cdot}5}=\right)$ 3 pounds of coal per hour. It may well be doubted if it was ever really obtained for even this in any steam-engine, the numerous published statements of interested parties and incompetent witnesses to the contrary notwithstanding.

15th. Assuming the cylinder initial total pressure to be unity, then the total average pressure, when cutting off at 0·19 of the stroke of the piston, was, for a mean of all the experiments, 0·5685. This is very nearly what it should be according to the Mariotte curve, including the effect of the steam in the cylinder passage and clearance.

Notwithstanding this coincidence, however, for the total *average* pressure during the stroke of the piston, the pressure in the cylinder at the end of the stroke averaged, for all the experiments, one-third of the pressure at the point of cutting off, instead of a little over one-fifth. Of course, this excess of pressure during the latter part of the expansion portion of the stroke, must have been compensated by a corresponding deficiency of pressure during the first part, showing that the steam in the cylinder during the early part of the expansion underwent an enormous condensation, replaced during the latter part by an equally large re-evaporation, the mean not differing sensibly from that due to the Mariotte law.

Again, this result, though most marked when saturated steam was used in the cylinder with air in the cylinder-jacket, was likewise strongly marked even during the experiments when the steam was adheated by the coil and used with steam in the cylinder-jacket; showing that, notwithstanding all the precautions

to prevent condensation in the cylinder, yet so great was the tendency of the steam to condense by its own expansion, that during the first part of the expanding a large condensation took place, the water of which was wholly re-vaporized before the end of the stroke.

When cutting off at 0·83 of the stroke of the piston, the final pressure in the cylinder was just what was due to the point of cutting off by the Mariotte curve including the effect of the steam in the cylinder passage and clearance, but the total average pressure was less than according to that curve.

16th. The economic gain derived from the higher measure of expansion employed under the various conditions with which the steam was used, will be found in the following table. The headings of the columns are so full that but little additional explanation is required.

TABLE EXHIBITING THE GAIN DUE TO CUTTING OFF THE STEAM AT 0·19 OVER THAT DUE TO CUTTING IT OFF AT 0·83 OF THE STROKE OF THE PISTON FROM THE COMMENCEMENT, FOR THE VARIOUS CONDITIONS GIVEN IN THE HEADINGS OF THE COLUMNS AND UNDER WHICH THE STEAM WAS USED.

| | | COST OF THE TOTAL INDICATED HORSE POWER, | | | | | |
|---|---|---|---|---|---|---|---|
| | | Inclusive of Condensation in the Steam-Jackets and Tubular Adheater. | | | Exclusive of Condensation in the Steam-Jackets and Tubular Adheater. | | |
| | | Pounds of Steam per hour when cut off at 0·19. | Pounds of Steam per hour when cut off at 0·83. | Gain by cutting off at 0·19 in per cent. of the cost when cutting off at 0·83. | Pounds of Steam per hour when cut off at 0·19. | Pounds of Steam per hour when cut off at 0·83. | Gain by cutting off at 0·19 in per cent. of the cost when cutting off at 0·83. |
| | | A | B | C | D | E | F |
| Saturated Steam with Air in Cylinder-Jacket, | | 67·15653 | 79·42902 | 15·45 | . . | . . | . . |
| Steam Adheated in Coil, | Steam in Cylinder-Jacket, | 37·15161 | 42·66123 | 12·91 | 25·83281 | 34·11702 | 24·28 |
| Steam Adheated in Tubular Adheater, | Steam in Cylinder-Jacket, | 40·86778 | 46·79664 | 12·67 | 28·20261 | 37·36986 | 24·53 |
| Steam Adheated in Steam-Pipe, | Steam in Cylinder-Jacket, | 39·97301 | 47·14389 | 15·21 | 28·08513 | 38·75460 | 27·53 |
| Means, | | | | 13·60 | | | 25·45 |

(TABLE CONTINUED.)

| | | COST OF THE NET INDICATED HORSE POWER ON THE SUPPOSITION OF AN EQUAL TOTAL INITIAL PRESSURE IN THE CYLINDER OF 35 POUNDS PER SQUARE INCH, AND AN AGGREGATE BACK AND FRICTION PRESSURE OF 5 POUNDS PER SQUARE INCH OF PISTON, | | | | | |
|---|---|---|---|---|---|---|---|
| | | Inclusive of Condensation in the Steam-Jackets and Tubular Adheater. | | | Exclusive of Condensation in the Steam-Jackets and Tubular Adheater. | | |
| | | Pounds of Steam per hour when cut off at 0·19. | Pounds of Steam per hour when cut off at 0·83. | Gain by cutting off at 0·19 in per cent. of the cost when cutting off at 0·83. | Pounds of Steam per hour when cut off at 0·19. | Pounds of Steam per hour when cut off at 0·83. | Gain by cutting off at 0·19 in per cent. of the cost when cutting off at 0·83. |
| | | G | H | I | J | K | L |
| Saturated Steam with Air in Cylinder-Jacket, | | 89·69607 | 93·28144 | 8·84 | . . | . . | . . |
| Steam Adheated in Coil, | Steam in Cylinder-Jacket, | 49·62090 | 50·10135 | 0·96 | 34·50300 | 40·06703 | 18·88 |
| Steam Adheated in Tubular Adheater, | Steam in Cylinder-Jacket, | 54·58409 | 54·95797 | 0·68 | 37·66817 | 43·88716 | 14·17 |
| Steam Adheated in Steam-Pipe, | Steam in Cylinder-Jacket, | 53·38903 | 55·36578 | 3·57 | 37·51126 | 45·51340 | 17·58 |
| Means, | | | | 2·26 | | | 15·21 |

Columns A and B contain the cost of the total indicated horse power *inclusive* of all condensations, internal and external, when cutting off respectively at 0·19 and at 0·83 of the stroke of the piston from the commencement. These are also the true relative practical values for equal developments of power in equal times by the same engine overcoming the same load with equal net effective, back and friction pressures. The quantities are the means of all the experiments with the exception of those on the first line for "saturated steam with air in the cylinder-jacket," which is the mean of the experiments when the cylinder was felted inside the jacket, so as to be under proper practical conditions for comparison.

The quantities in column C are the per centum which the difference of the quantities in columns A and B is of the quantities in column B. They show the saving under the above conditions of equal pressures, &c., that has been effected by increasing the measure of expansion from that due to cutting off at 0·83 to that due to cutting off at 0·19 of the stroke. The average gain is 13·60 per centum, and seems to have been but little influenced by the steam-jacketing and adheating.

Columns D and E contain the cost of the total indicated horse power *exclusive* of the condensation in the steam-jackets and tubular adheater, when cutting off respectively at 0·19 and at 0·83 of the stroke of the piston from the commencement. These quantities are, of course, not commercial or true economic values, but it is useful to know the gain that would be realized were there no cost attending the realization. They show, under these conditions, the relative values for equal developments of power in equal times by the same engine overcoming the same load with equal net effective, back and friction pressures.

The quantities in column F are the per centum which the difference of the quantities in columns D and E is of the quantities in column E. They show the saving, under the above conditions, which has been effected by increasing the measure of expansion from that due to cutting off at 0·83 to that due to cutting off at 0·19 of the stroke. The average gain is 25·45 per centum, or in round numbers twice the gain that was derived when all the condensations are included in the cost of the power.

Columns G and H contain the cost of the net indicated horse power under true comparable practical conditions, and give correct commercial relative values of the cost of the power when cutting off the steam at 0·19 and at 0·83 of the stroke of the piston from the commencement.

By correct comparable practical conditions are meant the maximum cylinder initial pressure maintained in practice with condensing engines, which is taken to be 35 pounds per square inch of piston above zero; and the minimum back and friction pressures, the sum of which is taken to be 5 pounds per square inch of piston above zero. The correct measure of the power is taken to be the net indicated horse power, and the cost the total weight of steam evaporated in the boiler.

The quantities contained in column I are the per centum which the difference of the quantities in columns G and H is of the quantities in column H. They show the saving, under the above conditions, which has been effected by increasing the measure of expansion from that due to cutting off at 0·83 to that due to cutting off at 0·19 of the stroke. The average gain is only 2·26 per centum. This is for the most favorable practical conditions; with the average conditions of practice the gain would be reversed.

The above gain of 2·26 per centum is in *fuel* only. To obtain it when developing equal powers in equal times, the sizes of the engine must be as 0·5685 when cutting off at 0·83 is to 0·9620 when cutting off at 0·19; in other words, with the higher measure of expansion the engine must be $\left(\frac{0{\cdot}9620 - 0{\cdot}5685 \times 100}{0{\cdot}5685} = \right)$ 41 per centum larger, and consequently 41 per centum heavier and more costly. This would overbalance in a commercial view a far greater gain in fuel than the dubious one of 2·26 per centum. The practical

engineer will appreciate, too, that had the point of cutting off been at about ⅔ds of the stroke instead of at 0·83, the 2·26 per centum gain would have been more than reversed.

From this it will be perceived that, so far as the problem can be solved by these experiments, it seems utterly futile to expect any increase of commercial economy in steam power, by using the steam with a higher measure of expansion than that due to cutting off at about ⅔ds of the stroke of the piston from the commencement, even when the most efficient steam-jacketing and adheating are employed.

Columns J and K contain the cost of the net indicated horse power under the same conditions as in columns G and H, but exclusive of the condensation in the cylinder-jackets and tubular adheater, when cutting off at 0·19 and at 0·83 of the stroke of the piston. These quantities have no commercial use, but they serve to show that under true comparable practical conditions, the gain due to the higher measure of expansion, even exclusive of the condensation in the cylinder-jackets and tubular adheater by which it was obtained and which is an inseparable accompaniment, was only 15·21 per centum (mean of column L).

17th. That the condensation in the cylinder, exclusive of that due to the production of the power, remains sensible even under the most efficient conditions for its prevention, and, under equally efficient conditions, is greater with the higher measure of expansion.

The following table exhibits the influence upon the condensation exerted by the different methods of using the steam.

| | Per centum of the Steam which entered the Cylinder that remained condensed at the end of the Stroke of its Piston, exclusive of the condensation due to the production of the power, and notwithstanding the re-evaporation on the Steam side of the Piston. | |
|---|---|---|
| | Steam cut off at 0·19. | Steam cut off at 0·83. |
| Saturated Steam with Air in Cylinder-Jacket, . . . . | 60·48 | 59·22 |
| Adheated Steam with Air in Cylinder-Jacket, . . . . | 58·96 | . . |
| Saturated Steam with Steam in Cylinder-Jacket, . . . . | 33·32 | . . |
| Steam Adheated in Coil with Steam in Cylinder-Jacket, . . | 11·28 | 0·55 |
| Steam Adheated in Tubular Adheater with Steam in Cylinder-Jacket, . | 22·30 | 12·76 |
| Steam Adheated in Steam-Pipe with Steam in Cylinder-Jacket, . | 21·91 | 13·51 |

18th. The following table exhibits the states in which the steam that entered the cylinder entered the condenser.

| | Per cent. of the Steam that entered the Cylinder which left it in the vaporous form at the end of the stroke of its piston. | | Per cent. of the Steam that entered the Cylinder which left it in the vaporous form during the time the exhaust passage remained open. | | Per cent. of the Steam that entered the Cylinder which left it in the form of finely divided water or spray suspended among the Steam. | |
|---|---|---|---|---|---|---|
| | Cut off 0·19. | Cut off 0·83. | Cut off 0·19. | Cut off 0·83. | Cut off 0·19. | Cut off 0·83. |
| Saturated Steam with Air in Cylinder-Jacket, . . | 35·40 | 87·20 | 54·25 | 53·87 | 10·35 | 8·93 |
| Adheated Steam with Air in Cylinder-Jacket, . . | 37·23 | . . | 51·55 | . . | 11·22 | . . |
| Saturated Steam with Steam in Cylinder-Jacket, . . | 59·73 | . . | 29·20 | . . | 11·07 | . . |
| Steam Adheated in Coil with Steam in Cylinder-Jacket, . | 78·70 | 91·46 | 0·41 | 0·02 | 20·89 | 8·52 |
| Steam Adheated in Tubular Adheater with Steam in Cylinder-Jacket, . . . . . . | 67·60 | 80·12 | 0·28 | –0·29 | 32·12 | 20·17 |
| Steam Adheated in Steam-Pipe with Steam in Cylinder-Jacket, | 68·76 | 79·61 | –1·46 | 2·58 | 32·70 | 17·81 |

In the above table, the quantities are the means from all the experiments. Those in the first two columns are the per centum which the quantities on line 52 are of those on line 4 minus the sum of those on lines 5 and 6, Table No. 1.

The quantities in the next two columns are the remainders after deducting those in the first two columns from those on line 38, Table No. 2.

The quantities in the last two columns are the remainders after deducting those on line 38, Table No. 2, from 100· or unity.

19th. The coal used in all the experiments was the same. It was from the Trevorton mines, and is known to be unsurpassed by any in steam generating efficiency. The combustion was slow, and the firing performed with the greatest regularity and care. Under these highly favorable conditions the evaporation must be regarded as a maximum for the type and proportion of boiler employed. The following are the total quantities and mean results of all the experiments; the weight of water being by tank measurement.

| | |
|---|---|
| Total number of hours, | 1,337. |
| " pounds of coal consumed, | 10,939. |
| " " of refuse in ashes, clinker, and dust, | 1,650. |
| " " of combustible, | 9,289. |
| Per centum of refuse from the coal, | 15·09 |
| Mean area of grate surface in square feet, | 1·4707 |
| Pounds of coal consumed per hour per square foot of grate surface, | 5·563 |
| " of combustible consumed per hour per square foot of grate surface, | 4·724 |
| " of water evaporated from a temperature of 100° Fahr. by one pound of coal, | 8·28909 |
| " " " " " " " " of combustible, | 9·41212 |
| " " " " " of 212° " " of coal, | 9·77363 |
| " " " " " " " " of combustible, | 11·08399 |

In general, the maximum variation in the evaporation per pound of combustible under sensibly equal conditions was $7\frac{1}{2}$ per centum of the greater quantity.

Also, that an increase of 50 per centum in the grate surface, *provided equal quantities of coal were burned in the same time*, produced no appreciable effect on the economic evaporation.

Finally, that each increase in the rate of the combustion was attended with a decrease in the economic evaporation, and in sensibly the following proportion, the coal being in pounds per hour per square foot of grate surface, and the water being in pounds per pound of coal from a temperature of 100° Fahr.

| COAL. | WATER. |
|---|---|
| 3 | 8·7 |
| $4\frac{3}{4}$ | 8·5 |
| $6\frac{1}{2}$ | 8·1 |
| 9 | 7·6 |

# THE PUMPING ENGINE

OF THE

# BROOKLYN WATER WORKS.

H

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THE results, described in the preceding paper, attending the employment of a steam-jacket with WATERMANN'S small experimental engine, were so great that it became of the first importance to ascertain to what extent they could be realized on large engines working under the conditions of actual practice. For, of course, we knew perfectly well that the results from the small engine could not be obtained in the same degree from a large one,—though doubtless they would in kind,—owing to the fact that in so small an engine the surface and weight of the cylinder, compared with its capacity, was enormously greater than in a large engine, and that the value of the jacket was some function of the relation between the surface and weight, and the capacity of the cylinder.

Under these circumstances it was determined to make an experiment on the Brooklyn Pumping Engine at Ridgewood, which had a first class cylinder of 90 inches diameter and 10 feet stroke of piston; steam-jacketed upon the sides, but not upon the ends. The object was merely to ascertain the absolute condensation in such a cylinder, working first with the steam admitted to the jacket, and then with the steam excluded from the jacket. This would determine the absolute amount of possible gain by the prevention of condensation, and also the extent to which such a jacket would prevent it. In making this experiment advantage was taken of the opportunity it afforded to ascertain the whole performance of the engine and boilers, and believing the facts discovered to be of general value, I have elaborated my notes into the present paper, in which will be found the experimental results, the manner of making the experiments, and a description of the machinery in as much detail as is required for the purposes of the practical engineer.

The engine at Ridgewood, Long Island, on which these experiments were made, belongs to the Municipality of Brooklyn, and is employed in supplying that city with water from a number of connected ponds. It is situated about six miles from Brooklyn in a spacious and handsome brick building, and elevates the water 160 feet into a large reservoir whence it flows to the city through pipes. The main, connecting the engine and reservoir, is of cast iron, 3 feet feet in diameter and 3450 feet in length, and is laid in a straight line with but one vertical curve of 800 feet radius. The machinery was constructed at the Works of Messrs. WOODRUFF & BEACH, Hartford, Conn., from the designs of their superintendent Mr. WILLIAM WRIGHT, and has been in regular operation since January 24th, 1860, pumping, with the exception of Sundays,

about eleven hours out of each twenty-four. The entire weight of the machinery, including engine, pumps, boilers, and all appurtenances, is four hundred and forty tons.

In making these experiments every facility was afforded by the water works officials.

The dimensions of the machinery are taken from the original drawings, and are as follows:—

## ENGINE.

The engine is a condensing one, and consists of one vertical, double-acting cylinder, steam-jacketed on the sides but not on the ends, which are, however, hollow and filled with powdered charcoal for a non-conductor. The piston actuates a cast iron beam weighing 25 tons, supported overhead by cast iron standards, on each side, bolted to the bed-plate. From this beam the motions are taken for the engine air-pump, and for the two water-pumps. The cylinder and standards rest upon and are bolted to a strong cast iron bed-plate extending the whole length of the engine, and secured to the masonry of the foundations. The boxes of the main centre of the beam are of cast iron, lined with Babbit's metal. The air-pump is vertical and double-acting; it has a stroke of 5 feet, and is worked from a journal half way between the main centre of the beam and its end centre, opposite the cylinder. At the same distance from the main centre towards the cylinder is a similar journal for working a feed-pump; but owing to some necessary modifications made on the original plan of valve-gear, and which required the space appropriated to this pump, it was dispensed with, and the boilers are fed by a separate small steam-pump of 6 inches diameter of cylinder and 9 inches stroke of piston. The condenser is a cast iron cylinder of 4 feet internal diameter, with a dome top, in the centre of which the exhaust steam is discharged. The air-pump and condenser are bolted, side by side, to a channel-plate containing a foot-valve, and situated below the bed-plate of the engine. This channel-plate is bolted, independently, to the stone foundation. The bottoms of the condenser and air-pump are upon the same level, but the top of the condenser rises above the top of the pump, so as to allow an upper horizontal channel way, parallel to the lower one, and containing a foot-valve also to connect the top of the pump with the condenser. A solid plate, with raised ledge, is placed within the condenser just below the upper channel way; it extends half across the condenser and intercepts half the injection-water which is drawn off into the upper part of the pump, while the remaining half is drawn off into the lower part. The pump-piston is solid and packed in the usual manner with hemp. The air-pump foot-valves are of gum, and seat upon brass grillages. The lower delivery-valve is the same, but the upper delivery-valve is in the form of a floating top, or large disc-valve, surrounding the pump piston-rod which acts as a guide for it, and seating upon wood let in a groove in the top of the pump-barrel. This barrel is of cast iron lined with brass. The injection-water is obtained from the same well that supplies the water pumps, and into this well it is again returned after passing through the condenser and air-pump, so that after being used for condensation it is carried to the reservoir by the water-pumps.

The steam piston is of cast iron, ribbed and hollow, and is packed with cast iron rings set out by steel springs: there are two rings, one behind the other. The cylinder-valves are of the usual double poppet kind, balanced, one for the steam and one for the exhaust, and as the cylinder is double-acting, a similar valve-chest and valves are required at each end; the two chests are connected in the usual manner with a vertical steam and exhaust side-pipe of 20 inches diameter. The side-pipes, valve-chests, and exterior of steam-jacket, are protected with a thick coat of felt covered by a handsome black walnut lagging. The steam is cut off by the steam-valve, which is made to act as an expansion-valve also, by means of a momentarily variable tripping apparatus that detaches and drops the valve by its gravity, while it is prevented from slamming into its seat by a dash-pot arrangement of air-cylinder and piston attached to it.

The lift of the steam-valve is only ⅝-inch; the lift of the exhaust-valve is 4 inches. Both valves are raised and seated quickly, and the exhaust-valve is held nearly at its full lift during a considerable part of the stroke by the cam-like action of its gear. The valve-gear is operated by the beam, and by a small brass cylinder of 10 inches diameter having receiving and delivering-valves and being supplied with water pressure from the main. It is, in effect, a variation of the cataract, and causes the engine to make a pause between each stroke of piston. The steam and exhaust-valves open and close precisely at the end of the stroke of the piston. The mechanical details of the valve-gear are very complicated, and it was a very costly piece of mechanism. Though automatic, it requires constant attention from the engineer, as the slightest variation either in the boiler pressure or in the vacuum must be corrected by the throttle. If this correction be not made, either the buffers attached to the piston rods of both pumps, will strike the guard timbers, or else the piston will shorten its stroke; according to the direction of the variation. The first endangers the machine, the second wastes steam in the increased clearance given to the steam-piston, for there is no cushioning. The demand for unceasing attention is a great defect in this gear, and the practical result is that the stroke of the pistons of the steam-cylinder and pumps instead of being 10 feet as designed, only averages 9⅔ feet.

The water-pumps (more particularly described under that head) are two in number; they are placed at opposite extremities of the beam, and at different elevations. The lower pump is placed immediately beneath the steam-cylinder. The pistons of both pump and cylinder are attached to the same piston rod which passes through stuffing-boxes in both ends of the cylinder, and through a stuffing-box in the pump-cover. The diameter of this rod from the cross-head to the bottom of the steam-piston is 9 inches, there a square shoulder is made and thence to the bottom of the pump-piston the diameter is 8½ inches. Between the cylinder and the pump, and upon the piston-rod, is placed a cast iron weight, of about 10 tons, carrying a buffer beneath which is a cob-work of timbers suspended from the bed-plate of the engine for preventing the piston of the steam-cylinder from striking the cylinder ends when the steam load is too great for the water load; this weight also supplies inertia at the commencement of the stroke of the piston. The upper pump is placed at the opposite end of the beam, and upon its piston-rod, between the pump and the beam, is placed a cylindrical cast iron weight of about 20 tons to supply inertia at the commencement of the stroke, and to counterbalance the opposite weight, the steam-piston, rod, &c. This weight is provided with buffers similar to those on the other rod and for the same purpose; and the timber cob-work placed beneath the buffers for arresting the stroke is similarly made to the other, but is supported on the bed-plate of the engine.

The water pumps deliver into a large cast iron air-vessel, cylindrical in form, with a domed top. It is 19 feet in extreme height above the top of the entrance nozzle, and 6½ feet in internal diameter. Its purpose is to secure by the elasticity of the compressed air within it, a uniform movement of the water through the main. As there are two joints above the water line for the air to leak out, it was found necessary to provide a small pump for supplying the air leakage. This pump is 4 inches in diameter by 5 feet stroke of piston: it is single-acting, and is worked from a projection on the air-pump piston rod.

Between the pump and the air-vessel, there was placed a cast iron double-seated check-valve of 3 feet diameter. This valve was expected to slide horizontally on a spindle, but its weight was so great and its area so large, that the available pressure was insufficient to close it, and it rusted fast upon its spindle. As a succedaneum, after the completion of the engine, a valved diaphragm was placed across the air-vessel above the receiving nozzle. It is fitted in the centre with one double-seated cast iron valve of 24½ inches diameter opening upwards; and a number of smaller valves surrounding it, but opening

downwards, and having an aggregate area less than that of the central one. The object of this arrangement is to gain time at the end of the stroke for the gentle closing of the pump-valves, for the reduced area of the diaphragm-valves opening downwards prevents the air compressed above them from returning too quickly upon the pump-valves, as the rush of water through them slackens. The slamming of the pump-valves is thus prevented.

The vertical movement of the cylinder and pump piston-rods is directed by an elegant arrangement of parallel motions.

The air-vessel is at the extremity of the engine opposite the cylinder. A handsome entablature extending horizontally the length of the engine at the level of the top of the standards is supported at one end from the cylinder and at the other from the air-vessel, the centre being sustained by the standards on which rest the pillow-blocks of the main journals of the beam. A commodious gallery projects from the top of the entablature and gives access to the beam and other upper journals. The fixed journals of the radius bars of the parallel motions are also supported by the entablature.

In the original design of the engine, it was intended to carry the steam in the boiler at a pressure of from 25 to 30 pounds per square inch above the atmosphere and to cut it off very short, expanding about eight times; the cylinder was, therefore, made large enough to give the proper average pressure for that measure of expansion; but upon the first trial it was soon ascertained that the engine could not be worked with a greater initial pressure on the piston than a few pounds per square inch above the atmosphere, and that instead of cutting off the steam at one-eighth the stroke of the piston or at 15 inches from its commencement, it was necessary to cut it off at six-tenths of the stroke or at 6 feet from the commencement. Thus not only was all the imagined benefit from large expansion lost, but there were realized all the serious disadvantages of using a cylinder two and three-quarters times too large for the work it had to perform. By using an initial steam pressure in the cylinder of 25 pounds per square inch above the atmosphere and cutting it off at six-tenths of the stroke of piston, the work now done by the 90 inches, diameter cylinder, with a stroke of 10 feet, would have been performed by a cylinder of the same stroke of piston, but with only 55 inches diameter. The saving would not have been in the first cost alone, but equally in the after economy; for as the fricton and back pressures would have been greatly reduced in per centum of the total average pressure; and as the absolute friction and condensation of steam by the cylinder, developing equal power, would have been less, the duty would have been materially increased.

The great oversight committed, was the failure to discern the impossibility of using steam with much expansion in the case of a pumping engine, pumping by the steam direct, and unprovided with a large mass of matter on the steam side to be put in motion at the commencement of the stroke of piston and brought to rest at the end of it. If we suppose the matter (other than the water) set in motion by the engine to have no weight, and the movement of the watery column to be uniform, then the steam-pressure on the piston at every point of the stroke would have to remain constant in order to exactly balance the water-load whose resistance is constant and unaffected by speed. In fact, on this hypothesis, it would be impossible to either increase or decrease the steam pressure above this equilibrium; for the supply of more steam would only accelerate the speed of piston, without increasing the pressure on it, and a decrease of the pressure on the piston by closing the communication with the boiler, would bring it quickly to rest. Under these conditions it would be impracticable to at all use the steam expansively. But just in proportion as we add matter on the steam side, can we increase the initial pressure on the steam-piston above an equilibrium with the water-load, for as we have to give movement to this matter in addition to the

water-load, and thus endow it with momentum, we can close the communication between the boiler and cylinder, and allow the steam to expand as far below the pressure equilibrating the water-load as the momentum of the matter can supplement, until we reach the point where the combined steam pressure and momentum are in eqilibrium with the water load. In a pumping engine, therefore, the maximum degree of expansion is limited by the momentum of the matter set in motion; the greater this momentum, the more expansively can the steam be used. In a word, we are enabled to use the steam expansively only by availing ourselves of the inertia of matter at the commencement of the stroke of the piston, and of its momentum at the end.

In the Cornish engines employed for pumping out mines, the large weight of matter required to give, in conjunction with the piston's speed, the necessary momentum for expansions of even three and four times, is obtained from the great length of pump-rod employed—extending from the surface of the ground to the bottom of the mine. If the depth of the mine does not furnish the weight for the desired expansion, it must be obtained by adding it for that special purpose. But in the design of the Brooklyn pumping-engine this essential provision was ignored, and an expansion of eight times was intended with conditions that absolutely prohibited the employment of any expansion whatever. The consequence was, as might easily have been predicted, that when put in operation it presented the anomaly of an engine fitted with a momentarily variable expansion-gear, from which great economy was anticipated, using its steam *ex necessitate*, almost without expansion. This defect, after being practically developed, was attempted to be made good by the addition of about eighteen tons of cast iron in the circumference of two semicircles of 14½ feet extreme diameter, keyed upon a shaft receiving a vibratory movement from the piston-rod between the steam-cylinder and lower pump. These semicircles were so poised that the diameter would approach the horizontal at the half stroke, and the vertical at the end of the stroke, in order to give, beside their momentum, the greatest possible leverage at the beginning and end of the stroke,—the first for increasing the initial steam pressure in the cylinder, and the last for compensating the decreased steam pressure by the expansion. These vibrating segments mainly perform the function of a fly-wheel, but in a very inferior manner; for any superfluous momentum that may exist in the wheel at the end of the stroke of the piston, passes on and is utilized during the next stroke; but whatever *vis viva* the vibrating segments may possess at the end of one stroke, instead of being utilized during the next, is worse than lost; for it is expended in producing an injurious shock upon the engine. Even with the addition of the vibrating segments, the initial cylinder pressure cannot be raised above 6½ pounds per square inch of piston above the atmosphere, and the steam cannot be cut off shorter than six-tenths of the stroke of the piston from the commencement, allowing it to expand through the remaining four-tenths after having been throttled down to 1 pound per square inch above the atmosphere at the point of cutting off. The difference between the initial and final pressures is only about 11 pounds per square inch of piston, and this absolute quantity is the value of the aggregate *vis viva* of the whole system. The engine must work under these conditions of cylinder pressure precisely, or it cannot work at all.

Again, with the Cornish system, in which the steam acts indirectly by first raising the mass of matter whose descent afterwards performs the pumping, it is of no importance that the speed of the steam-stroke is both very rapid and very irregular, being greatest during the first part of the stroke, and least during the last part; for no injurious practical result will follow from raising this mass with great velocity or with great variations of velocity; but when the steam is applied to pump direct, the practical requirements are entirely changed; for it is essential that the water be started *very* slowly from its state of rest, and that any increase of velocity afterwards given, be bestowed by uniform accelerations. The descent of a weight by the

force of gravity admirably satisfies these conditions; and the problem with the Cornish system is simply that of two nearly equal weights, the greater by its descent lifting the lesser with a motion uniformly accelerated according to the laws of gravity, the force of which, however, is thus made to act with a very diminished effect as regards absolute velocity. In pumping, then, directly by the steam, it is practically impossible to employ a high measure of expansion without a fly-wheel, crank, and air-vessel. The first to equalize the power throughout the stroke, the second to cause the piston to begin and end its stroke very gradually, and the last to neutralize the effect upon the water of the too great difference in the speed of the piston at the ends and middle of its stroke. Of these three essentials, the Brooklyn engine, pumping directly by the steam, possesses but the last, and it is probable that the use of more vibrating weight and a higher expansion would produce evils from irregularity of motion disproportioned to the benefit.

With the fly-wheel and crank arrangement above alluded to, it is, of course, not intended to pass the power through the shaft of the wheel. The pumping is to be done directly from the beam, and the crank is added for the purpose of measuring out the stroke exactly, of obtaining a rotary motion for the wheel, the sole function of which is to supply momentum, and to permit the use of the usual eccentric with its simple valve-gear.

Had the Brooklyn engine been fitted with crank and fly-wheel, there would have been saved 2 inches of each stroke of the piston in clearance; for with a positive measure of the length of the stroke, the clearance at each end of the cylinder need not have exceeded 1 inch, whereas the present *working* clearance is 3 inches.

Instead, too, of a costly, complex, and troublesome valve-gear, requiring the constant and vigilant attention of the engineer, as before stated, there might have been employed the simple and elegant eccentric, with its unequalled appropriateness of valve motion; and, finally, momentum could have been commanded for any measure of expansion desired.

The cost of such an arrangement would have been less than that of the present one; for the vibrating segments, their shaft and links, offset the fly-wheel, its shaft and connecting-rod, leaving the difference of cost of the valve-gears a clear gain. The whole system would have thus been rendered not only cheaper but simpler; for fewer parts would have been employed, and their action would have been more reliable, economical, and satisfactory.

## WATER-PUMPS.

The water-pumps (Plate IV.) are of peculiar construction. Each pump is composed of two concentric cylinders with a clear space of 7½ inches between them. The inner cylinder—36 inches in inside diameter and of 1½ inch thick cast iron—is the pump-barrel proper, and contains the piston which is fitted with one double-seated circular-valve, surrounding the piston-rod, opening upwards, and affording a net water-way of 2·1 square feet area or nearly 30 per centum of the area of the pump-barrel. This valve is composed of ½-inch thick boiler-plate, and seats upon pine wood placed endways in, and filling up, a groove 3 inches deep and 1¼ inch wide. One seat is 12½ inches above the level of the other. The packing of the piston is formed of two rings of 1 inch thick cast iron, placed one above the other, and having flanges at top and bottom so as to form a groove for the reception of the wood. Each ring is 5 inches deep and 2¾ inches wide; 1¾ inch of which, and between the flanges, is filled with lignum vitæ placed endways. The flanges, where they touch the pump-barrel, are hooped with brass, and serve merely to retain the wood. Each ring is in four segments which are pressed out against the pump-barrel in the usual manner by a hemp packing 1 inch thick. This packing is retained by a cast iron follower screwed up from beneath,—a very inconvenient

arrangement. The piston is secured to its rod by a head on the latter below the piston, and by a key through the rod and piston.

The outer cylinder is 54 inches in diameter, and is made of 1¼ inch thick cast iron. The annular space between the two cylinders is closed at the top by one double-seated circular-valve having one seat upon the top of the inner cylinder and the other upon the top of the outer one: the latter seat is 19 inches below the former. The seats are formed of pine wood let into grooves in the cast iron in the same manner as for those of the piston-valve. The net area of the annular space is 6·4 square feet, which is in addition to the 2·1 square feet area in the piston for the passage of water. The two cylinders are united by six arms cast on the inner one and with T ends; these arms extend across the annular space, and their ends are bolted to the outer cylinder. The valve is similar to the one in the piston, and, like that, is formed of ½-inch thick boiler iron. It has guides on the inner cylinder and also on the piston-rod, the latter being made of iron rods bowed over. The pump-barrel is not lined with brass,—an important omission.

The valves and seats just described are not the original ones. The first valves, though of sensibly the same form, were of cast iron and very heavy, requiring considerable pressure to lift them. They did not work in a satisfactory manner and were replaced with those of boiler-plate made as light as possible. In the first seats, the wood was faced with gum, which, for want of sufficient fastening, was liable to be pulled off by the valve on lifting; it was, therefore, removed and the valve allowed to seat on the wood.

The water-pumps are two in number; they are vertical, single-acting, and lifting; but receive the water at both ends of the barrel. At the lower end it enters through the double-seated valve in the piston. At the upper end it enters through the double-seated valve covering the annular space which surrounds the barrel and forms the channel-way for the water to reach the valve. On the descent of the piston both valves open, and upon its ascent they close; the annular arrangement being merely a contrivance to obtain with a lifting-pump an area of valve opening larger than the area of the pump-barrel. In the present case, the area of the pump-barrel is 7·068 square feet, and the aggregate area of opening is $(2{\cdot}1 + 6{\cdot}4 =)$ 8·500 square feet, and could have been made much larger by a slight increase in the diameter of the outer cylinder. During the descent of the piston the sole function of the pump is to fill; and it is only during the ascent that it discharges.

The two pumps are placed at different elevations, and at opposite ends of the beam. A closed top of cast iron, 2 inches thick, and having a delivery nozzle of 3 feet diameter, is bolted by flanges to the upper part of the outer cylinder of both pumps. This top is cylindrical, its outside diameter is 72 inches, and its extreme height is 66 inches; access is had to it by a movable cover of cast iron 2 inches thick and ribbed, which contains the piston-rod stuffing-box. The lower part of the outer cylinder of the lower pump, is bolted by flanges to an open cast iron stool with legs, which supports the pump in the well. The lower part of the outer cylinder of the upper pump is bolted by flanges to a closed cylindrical box of cast iron, 1⅝ inch thick, and having a receiving nozzle of 3 feet diameter. This box is 63½ inches in outside diameter; its extreme height is 64 inches, and access is had to it through a manhole in the side.

The lower pump is 14 feet below the upper one; it is situated in the well, and is partly immersed in the water; it has no foot-valve, and discharges its water through the upper pump by means of a wrought iron horizontal pipe, 3 feet in diameter and 21¼ feet in length, connecting the top of the lower pump with the bottom of the upper one. By this arrangement, the lower pump and wrought iron connecting pipe act as suction pipe for the upper pump; while the upper pump and the connecting pipe act as delivery pipe for the lower pump. It will thus be seen that the two pumps alternately pump through each

other. The pumps deliver into a large air-vessel, without a check-valve to prevent the return of the pressure; the only valves used are those above described in the piston of the pump, and in the annular space surrounding the pump; unless the diaphragm valves in the air-vessel can be considered as check-valves forming part of the arrangement.

The object of this system of the one pump pumping through the other, was to communicate *continuous* movement to the column of water passing through the main; and consequently to prevent the concussions, shocks, and losses due to its alternate rest and motion. In effect, however, this end was very imperfectly attained; to have succeeded, would have required the piston of one pump to have commenced its delivery at the instant of time the other had finished, and the speed of the pistons to have been uniform from beginning to end of stroke. Neither was the case. The pistons began and ended their stroke with much less velocitiy than they moved with during the middle of it, and, owing to the construction of the valve-gear, there was a pause of about $1\frac{1}{4}$ second at the end of each stroke: now, as the engine made, say, 16 strokes of piston per minute, it was actually at rest $(16 \times 1\frac{1}{4} =)$ 20 seconds out of 60; and so far as the action of the pump was concerned, it is probable the column of water would have come to a state of complete rest after each stroke of the piston. That it did not do so, was due to its own momentum and to the compressed air in the large air-vessel into which the pumps delivered, and from which proceeded the main,—a cast iron pipe, 3 feet in diameter, and 3450 feet in length,—that conducted the water into the reservoir. There was a check-valve placed between the upper pump and the air-vessel; it was a heavy, cast iron valve with two seats, and, being supported on a horizontal spindle, had no tendency to close by gravity. It was always inoperative and had rusted fast to its spindle, consequently the air-vessel pressure rested constantly upon the pump-valves and followed down its piston. The action of this back pressure was so injurious that the valved diaphragm previously described was added within the air-vessel to shut off a portion of the back pressure on the first part of the return stroke. There was no stand-pipe, and the air-vessel was depended on for the smooth working of the machine.

The reason for adopting the above described pumping system, was the belief that the current of water, both in the suction pipe and in the main, would be made to flow continuously by the sole action of the pumps without either changing velocity or coming to rest. This anticipated benefit was not realized, but the disadvantages of the system were, and among them may be mentioned: 1st. That it required one pump to be placed a vertical distance of 14 feet above the other, and to have a suction-pipe about 30 feet in length with two right-angled bends. The disadvantage of this pipe was twofold; first, in the great facility it afforded for air leakage; second, in the moderate limit it imposed upon the speed of piston. With the depth of water in the well during the experiments, the engine could not be worked faster than, say, eight double strokes per minute; for with less time the water, having to traverse the long and crooked suction pipe, could not reach the upper pump in sufficient quantity to fill it solidly; and there resulted both a great pounding of the valves and loss of action in the pump. With an increased depth of water in the well, the engine could be worked at ten double strokes; and had the upper pump been immersed like the lower one, sixteen double strokes could have been made judiciously. The piston of the lower pump at the bottom of its stroke, was 3 feet above the bottom of the pump-well; and with 5 feet of water in the well, the valves of the upper pump slammed heavily when the piston made eight double strokes per minute. With 7 feet of water in the well and eight double strokes of piston, the upper pump worked satisfactorily; and with 9 feet of water in the well, ten double strokes per minute were practicable. The effect of a slight leak in the suction-pipe was to reduce the stroke of the steam-piston from 10 feet to $9\frac{1}{3}$ feet to prevent it from striking the cylinder cover; and if air in any considerable quantity was worked the

pump-valves gave notice of it by their slamming. The reduction of the stroke of piston was attended with an increase of the clearance at each end of the cylinder, equal to half the reduction; and a consequent greatly increased loss of duty, beside the loss due to the pumping of air instead of water.

2d. By the distribution of the total space displacement of pump-piston into two pumps at opposite extremities of the engine-beam, pumping through each other, and without a check-valve to shut off the pressure from the air-vessel, it follows that, should the valves of either pump break or hang up, the water-load would be thrown upon the other pump-piston and, acting in conjunction with the steam pressure on the steam-piston and being unbalanced by any resistance at the opposite extremity of the beam, the descending mass would strike with probably sufficient force to break down the engine. Even with the addition of a check-valve, as both it and the pump-valve might break or hang up at the same time, the chances for the accident described, though greatly diminished, are still formidable. If the pumps acted independently without pumping through each other, no part of the water-load, in the event of their valves breaking or hanging up, would act *in conjunction* with the steam pressure.

3d. The annular system of lifting-pump offers, by reason of its greater surface, much more resistance to the passage of the water through it than the common pump does. It also makes packing the piston a very troublesome and inconvenient operation; for the packer has to enter beneath the lower part of the pump and pack overhead, screwing up a heavy follower in the same disadvantageous position. In order to pack the lower pump, in the actual condition of things, the entire water in the aqueduct has to be drained off to give the packer access. If the water be shut off by a gate, the packer enters at the risk of his life; for should it flow in by any accident he would certainly be drowned. Obviously, the best pump—because the simplest, cheapest, and most convenient—is the plunger in common use, which is packed from the outside and has the advantage that any leak in the packing is at once visible, whereas the piston of the lifting pump may leak much and for a long time before it is discovered. Also the valves being in a compartment separate from the barrel, can be made with any required area of opening, and with every convenience of bonnets for examination from the outside; and as such pumps act independently of each other, any derangement of the one would not be complicated with the action of the other, and both pumps could have been immersed in the well.

In point, then, of practical simplicity and of theoretical excellence, cost of construction, and convenience of arrangement for management, examination, and repair, the system of lifting-pump with air-vessel adopted for the Brooklyn Water Works is inferior to the common arrangement of plunger-pump and stand-pipe.

It has been objected to this system of pump, that part of the water has to be raised twice through a portion of the lift, because all that part of the water which enters the pump through the annular space has to be first raised to the height of the valve on the top of that space, and then let fall into the barrel; consequently the distance from the top of the pump to the centre of gravity of the water which has entered the pump through the annular space, is the amount of lift lost with that water. These facts are true, but there remains to be added that the weight of water falling into the pump through the top valve assists the pumping by pressing down the pump-piston with precisely the same power (friction apart) that was required to elevate this water the distance it was let fall into the pump-barrel.

## DIMENSIONS OF THE ENGINE, AND OF THE WATER-PUMPS.

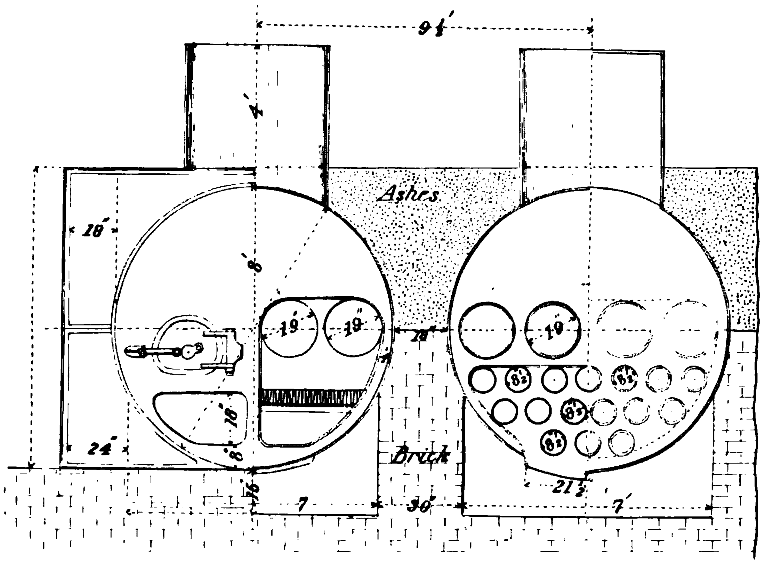
The following are the principal dimensions of the engine, and of the water-pumps. It will be observed that the stroke of the steam-piston (which determines the strokes of the water-pumps and air-pump, is given below at the *intended* figure of 10 feet; this is not the length of the stroke actually obtained when pumping. The working stroke is variable, but averages 9¾ feet. The calculations, too, of space displacement of pistons and of space comprised between the steam-piston at the end of its stroke and the exhaust-valve, are for the intended stroke of 10 feet, and not for the real one of 9¾ feet.

| | |
|---|---|
| Diameter of the cylinder, | 90 inches. |
| " piston-rod, | 9 " |
| Stroke of the piston (intended), | 10 feet. |
| Space displacement of the piston per stroke, exclusive of rod, (intended,) | 437·711 cubic feet. |
| Clearance (intended), | 1½ inch. |
| Diameters of the two discs of the steam-valve, | 14 and 14½ inches. |
| " " exhaust-valve, | 16 and 16½ " |
| Area of the steam-port, (8 by 45 inches), | 360 square inches. |
| Space comprised between the piston at the end of its stroke and the exhaust-valve (intended), | 13·467 cubic feet. |
| Capacity of the condenser, | 94·000 " |
| Diameter of air-pump (double-acting), | 36 inches. |
| " piston-rod, | 3½ " |
| Stroke of the air-pump piston (intended), | 5 feet. |
| Space displacement of the air-pump piston per stroke, exclusive of rod, (intended,) | 35·176 cubic feet. |
| Width of steam-jacket space, in clear, | 0⅞ inch. |
| Thickness of the metal of the cylinder, | 1⅝ " |
| " " steam-jacket, | 1¼ " |
| Total depth of steam-piston at edge, | 9 3/16 inches. |
| " " centre, | 18½ " |
| " cast iron packing rings for steam-piston, | 5½ " |
| Thickness of outer packing ring, | 1½ inch. |
| " inner packing ring, | 1 " |
| Extreme length of beam, | 32 feet. |
| Length of beam between axes of end journals, | 30 " |
| Depth of beam at centre, | 7⅛ " |
| Thickness of the web of the beam, | 6 inches. |
| Width of the flanges of the beam, | 13 " |
| Depth " " | 4½ " |
| Diameter of the main journals of the beam (one on each side), | 13 " |
| Length " " " | 24 " |
| Diameter of the end " " (one on each side), | 7½ " |
| Length " " " | 8½ " |

Plate III.

*Three Boilers for the Brooklyn*

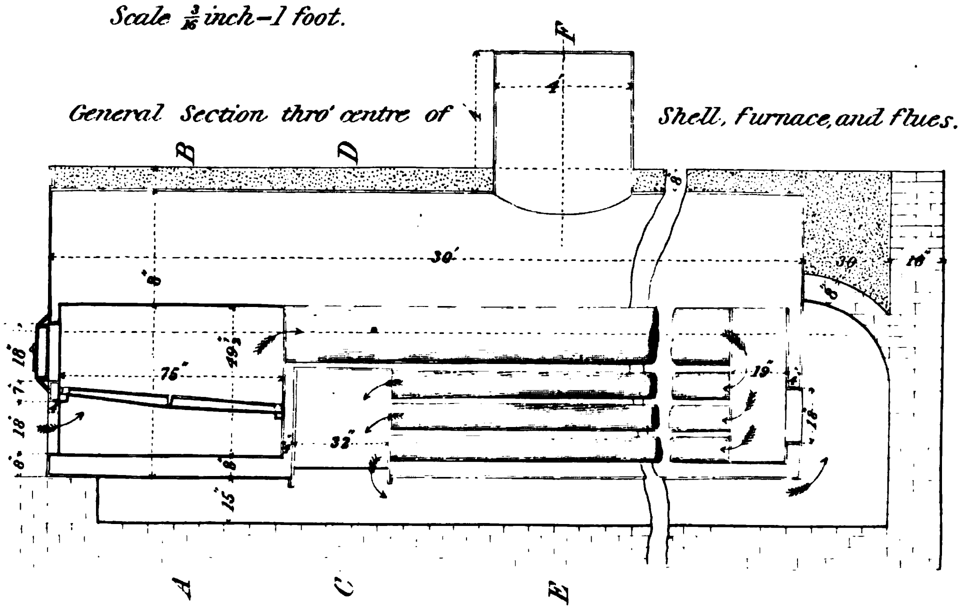
**PUMPING ENGINE.**




Elevation. *Section on AB.* *Section on CD.* *Secⁿ on EF.*

*Scale $\frac{3}{16}$ inch–1 foot.*

*General Section thro' centre of Shell, furnace, and flues.*



| | |
|---|---|
| Diameter of the links at neck (two links at each end of beam), | 6¼ inches. |
| " water-pumps (single-acting, lifting, and one at each end of beam), | 36 " |
| " water-pump piston-rod, | 8¼ " |
| Stroke of the water-pump piston (intended), | 10 feet. |
| Space displacement of the water-pump piston per stroke, exclusive of rod, (intended), | 66·974 cubic feet. |
| Area of opening for water through the water-pump piston, | 2·1 square feet. |
| " annular opening surrounding each water-pump, and for the passage of water, | 6·4 " |
| Total area of valve opening for each pump, | 8·5 " |
| Diameter of small air-pump for pumping air into air-vessel, | 4 inches. |
| Stroke " " " " (intended,) | 5 feet. |
| Extreme length of engine, over cylinder and air-vessel, | 48½ " |
| Extreme breadth of engine, | 8¾ " |
| Height from bottom of pump-well to bottom of cylinder, | 36¾ " |
| " cylinder to top of beam at half-stroke, | 29¾ " |

## BOILERS.—(Plate V.)

The boilers are three in number, placed side by side, in a setting of brick-work. They have single return drop-flues with a second return beneath the bottom of the shell. They all discharge their products of combustion into the same brick flue which delivers them into a brick chimney of 4 feet internal diameter and 100 feet height above the level of the grates. This brick flue passes across the rear ends of the boilers, so that the products of combustion from the boiler farthest from the chimney has to pass the outlets from the other two boilers; and the products of combustion from the centre boiler has, in the same manner, to pass the outlet from the boiler nearest the chimney.

The shell of each boiler is cylindrical with flat ends; it has a diameter of 8 feet and a length of 30 feet. Within the shell, and at one end of it, are placed the two furnaces and their ash-pits. Each furnace is 6½ feet long, with a 3 feet width of grate. The grate-bars extend the whole length of the furnace, the crown of which is 30 inches above them. From the crown of the furnace to the bottom of the ash-pit is 49½ inches. The inner sides of the furnaces and ash-pits are vertical, and the water-way between the furnaces is 5 inches wide. The outer sides of the furnaces and ash-pits conform to the shape of the shell, and have a water-way between them and the shell 4 inches wide. The top of the furnaces and bottom of the ash-pits are flat, with the angles of junction with the sides well rounded. The water-way between the end of the shell and the furnaces is 4 inches wide.

The grate-bars are of cast iron and in two lengths. Each bar is 1⅛ inch wide on top and 5/16-inch wide on bottom, depth at ends 2¾ inches, and at centre 4 inches. The top of the bars is channelled lengthways, and the air-space between them is ½-inch wide. The aggregate air-space between the grate-bars is, for one furnace, 5⅓ square feet.

The furnace door is of cast iron and semicircular, with the lower angles well rounded into the flat bottom; the extreme breadth of opening is 21 inches, extreme height 18 inches. The door has a hole of 3⅞ inches diameter to admit air, for the distribution of which there is a wrought iron lining ½-inch thick, perforated with two hundred and forty holes of ⅜-inch diameter on the outside and ¼-inch diameter on the inside. The ash-pits are fitted with doors containing registers.

From the back of each furnace two horizontal cylindrical flues, of 19 inches diameter and 21½ feet length, proceed to the back smoke connexion, which is a large flat-topped box extending across the boiler. The top of this box, the crown of the furnaces, and the top of the flues are upon the same level. The sides and bottom of the box conform to the shape of the shell, between which and it there is a water-way 4 inches wide. The width of this box, lengthwise the boiler, is 19 inches; and access is had to it for cleaning through a manhole, in the back end of the boiler, 18 inches in diameter.

From the lower part of the back smoke connexion of each boiler, sixteen horizontal cylindrical flues, 8½ inches in diameter and 18½ feet in length, proceed to the front smoke connexion. These flues lie underneath and parallel to the upper four 19 inches diameter flues. The width of water-way between the flues is 3½ inches.

The front smoke connexion is a flat-topped box extending across the boiler, and having sides conforming to the shape of the shell. The bottom of this box for a distance, crossways the boiler, of 3 feet 7 inches, and for its entire width of 2 feet 8 inches, lengthwise the boiler, is left open for the purpose of discharging the products of combustion into the lower flue formed by the bottom of the shell and by the brick-work of the setting. The forward smoke-box is immediately behind the ash-pit, between which and it there is a water-way 4 inches wide, and the same width of water-way is continued between the sides of the box and the shell. The products of combustion, after passing along the bottom of the shell of the boiler, are discharged into a brick flue common to all the boilers, and are thence delivered into the chimney.

It will thus be seen that the products of combustion, after leaving the two furnaces of each boiler, traverse first horizontally through the upper four 19 inches diameter flues, passing into the back smoke connexion whence they return horizontally to the front smoke connexion through sixteen 8½ inches diameter flues. From the front smoke connexion they descend beneath the bottom of the shell towards the chimney. They have thus changed their direction twice horizontally and twice vertically, and have traversed from the centre of the furnace to the point of exit from beneath the boiler, 71½ feet.

On the centre of the top of each shell is a cylindrical steam-drum 4 feet in diameter and 4 feet in height, with a flat top, on which are placed the steam-stop and safety-valves,—one to each boiler,—and from which the steam-pipe proceeds to the engine.

A brick wall, 18 inches thick at the horizontal diameter of the boilers, is placed between the boilers and at the outside of them. The two outside walls are carried up 8 inches above the tops of the shells: the two inside ones terminate at their horizontal diameter. The space above the shells and between the outer walls is filled with ashes. These four walls are of exactly the length of the boilers, the back ends of which and the steam-drums are thoroughly covered with felt. The same covering also surrounds the steam-pipes.

The flue beneath the bottom of each boiler, formed by the brick-work, is 7 feet in the clear, crossways the boiler, and of 15 inches depth at the centre of the boiler. The bottom is flat and the sides vertical.

In the cross brick flue common to the three boilers there is placed a heater composed of six wrought iron pipes, each 2½ inches inside diameter and 28 feet long, united at their ends by semicircular cast iron elbows, so as to make one continuous pipe. All the pipes lie in the same horizontal plane and form a continuation of the feed-pipe. The feed-water, after being taken from the hot-well, is pumped through this heater into the boilers, the water traversing the aggregate length, 168 feet, of the pipes. The object of this heater was, of course, to utilize a portion of the waste heat in the flue. There was but the one heater for the three boilers, and it presented to the hot gases an exterior surface of 121 square feet.

The following are the principal dimensions of the boilers, namely:—

| | |
|---|---|
| Number of boilers, | 3 |
| Extreme length of the shell of each boiler, | 30 feet. |
| " diameter " " | 8 " |
| Number of furnaces in the three boilers, | 6 |
| Length of grate-bars in each furnace, | 6 feet 3 inches. |
| Breadth " " | 3 feet. |
| Total grate surface of the six furnaces, | 112·5 square feet. |
| Aggregate heating surface in the furnaces of the three boilers, | 339·0 " |
| " " upper row of flues of the three boilers, | 1283·3 " |
| " " back smoke connexion " | 200·1 " |
| " " lower return flues " | 1976·0 " |
| " " front smoke connexion " | 161·3 " |
| " " bottom of the shell " | 558·3 " |
| Total heating surface in the three boilers, | 4518·0 " |
| Aggregate cross area of the upper row of flues of the three boilers, | 23·627 " |
| " " lower return flues " | 18·915 " |
| " " flue beneath the bottom of the shell of the three boilers, | 39·500 " |
| Cross area of the chimney of the three boilers, | 12·566 " |
| Height of the chimney above the level of the grates, | 100 feet. |
| Capacity of steam room in the three boilers and their steam-drums, | 1549 cubic feet. |

PROPORTIONS.

| | |
|---|---|
| Ratio of the heating to the grate surface, | 40·160 to 1·000. |
| " grate surface to the cross area of the upper row of flues, | 4·761 " |
| " " " lower return flues, | 5·948 " |
| " " " flue beneath the bottom of the shell, | 2·850 " |
| " " " chimney, | 8·952 " |

## MANNER OF MAKING THE EXPERIMENTS.

The experiments were directed mainly to the determination of the efficacy of the steam-jacket; for which purpose the engine was operated during the working hours of four days with the steam admitted to the jacket, and for the same length of time with the steam excluded from the jacket. As during the progress of the experiments, accurate measurements were taken of all the quantities developed, the data was obtained for determining the absolute power of the engine, the essential conditions of its working, its friction, general efficiency, and duty. Also the evaporation given by the boilers. The whole forming, in conjunction with the dimensions and a description of the machinery, reliable and valuable precedents for the practical engineer.

The load was absolutely uniform, for, consisting always of the lifting of the same weight of water per stroke of piston, it was unaffected, within reasonable limits, by changes in the velocity of the piston. The number of double strokes of piston made, then, is an absolute measure of the effect produced irrespective of time; and as this number was given by a counter moved by the engine itself, the work done in the two cases of the experiments was ascertained comparatively with perfect accuracy.

To determine the gross power developed by the engine in terms of the dynamical unit conventionally adopted by engineers, namely, the horse power of 33000 pounds raised one foot high per minute, two indicators were employed; one was placed at the top of the cylinder and the other at the bottom, their connecting pipes were screwed directly into the clearance of the cylinder, and were but a few inches in length. The indicators were large, fine instruments with piston area of half a square inch, and their springs were graduated to indicate one pound per square inch for each compression or extension of one-tenth of an inch. They were in excellent order, remained attached throughout the experiments, and always gave a perfect diagram. A diagram was taken every hour both from the top and from the bottom of the cylinder, and so uniform were they, that the lines of one coincided sensibly with the lines of all the others. From a collation of all the diagrams taken, there were determined the mean gross effective pressure on the piston during its stroke; the pressure in the cylinder at the commencement of the stroke,—at the point of cutting off the steam,—and at the end of the stroke. Also the back pressure against the piston, and the point of the stroke at which the steam was cut off.

The weight of water evaporated in the boiler was a most essential element of the experiments; it was ascertained by measuring the feed-water in a tank previously to pumping it into the boiler. This tank was of wood lined with lead and had the following dimensions, namely: height, 3 feet 7 inches; length, 6 feet 5½ inches; breadth, 7 feet; capacity, 162 cubic feet. The feed-water was first drawn from the hot-well and discharged into the tank, whence it was pumped into the boiler. The time when each tank was emptied was noted as a check upon the number.

It is evident that the weight of water, being the weight of steam produced in the boiler, is a perfect measure of the cost of the power when used under the same conditions; and is necessarily far more reliable for such a measure than the weight of coal consumed in the furnaces, especially in short experiments, as it is unaffected by the changes of weather, accident of firing, and inequality of constitution, which influence coal much, and render it not a sufficiently delicate means for the detection of small and feebly marked differences of economic result.

A strict examination of the boiler and all the pipes connected with it, made before commencing the experiments, failed to reveal any leakages either of water or steam.

Each of the three boilers was provided with a glass water gauge; and at the commencement of each experiment the height of the water in it was carefully noted together with the pressure of the steam: at the end of the experiments the height of the water and the pressure of the steam were left precisely as at the commencement. During the experiments the water level in the boiler was not allowed to fluctuate to any extent of practical importance, and the steam pressure in the boiler did not vary more than a pound and a-half per square inch.

As regards priming, not the slightest indication of it could be observed. The cylinder was very favorably located for obtaining dry steam, as its bottom was considerably above the top of the boilers, and the steam-pipe, which was about 50 feet long, delivered the steam into the lower valve-chest; to reach the upper valve-chest the steam had to ascend 10 feet higher. The auxiliary steam-pump (continually working) took its steam from the lowest point of the main steam-pipe, and drew off with it all the water in that pipe, due to any cause whatever, so that none entered the cylinder.

There were, also, other favorable conditions. The steam was throttled down from a pressure of 12 pounds per square inch above the atmosphere in the boilers to 6½ pounds above it in the cylinder at the commencement of the stroke, and to 1 pound above it at the point of cutting off, which was at six-tenths of the stroke from the commencement. The steam room in the boiler was thus virtually increased, and the velocity of

the steam current in the steam-pipe diminished. The steam, too, was taken from the boilers at the height of 6½ feet above the water level.

For ascertaining the boiler pressure, ALLEN'S spring gauges were employed, one for each boiler, and the mean of the three was noted hourly. The indications of a similar gauge, graduated to represent inches of mercury, and applied to the condenser, was noted hourly. The height of the barometer was taken at the same time, in order to furnish the correction for the total pressures by gauge and indicator.

The temperatures were regularly taken of the engine room, of the water in the pump-well, of the feed-water in the tank, and of the same when entering the boiler after having passed through the heater in the brick flue common to all the boilers. The temperature of the products of combustion after leaving the boiler was approximately ascertained by lowering into the flue just below its top, a piece of tin and a piece of lead suspended by a small iron wire, the opening through which they were inserted being tightly closed. On withdrawing them after the lapse of an hour, the tin was always found completely melted off, but the lead remained unchanged. Again, on withdrawing the wire a few minutes after inserting it, the tin was found all melted off as before. These trials were repeated several times with the same results, and establish the temperature of the heated gases in the flue at between the melting points of tin and lead, say, approximately, at 500° Fahr.

The weight of steam condensed in the steam-jacket was ascertained by drawing it continuously off into a tank as fast as formed into water, through a small pipe inserted in the bottom of the jacket, and fitted with a cock which was graduated so as to just discharge the water without steam. From time to time the cock was opened wide for a moment to be certain that the whole water was delivered, The temperature of this water was taken when each tank was filled, in order to correct for the weight by the capacity of the tank. The time of filling each tank was noted as a check upon the number. The capacity of the tank was exactly 5½ cubic feet. The communication between the boilers and jacket was through two small pipes fitted with screw-valves at both ends. One, inserted into the steam-pipe, filled the jacket with steam, entering several feet above the bottom; the other, inserted in the water-space of the boiler and the bottom of the jacket, allowed the water of condensation to flow back to the boiler. When the jacket is used under ordinary conditions of working, both pipes are left open, but during the experiments the lower one was closed. When, during the experiments, steam was excluded from the jacket, there was no appearance of any leakage past the valves of these pipes.

During the portion of the twenty-four hours that the engine was at rest, banked fires and steam were kept in the boilers, and steam was kept in the jacket, and, in fact, in the side pipes, valve-chests, and cylinder also; for in a short time a very slight leakage past the throttle and cylinder-valves would fill all; the consequence was that no fuel was lost in the commencement of an experiment to heat up the metal, and that the results obtained were purely normal and due to the actual working of the engine the same as though the experiments had been a middle portion of an indefinitely long one. The consumption of coal given in the table of the experiments is, of course, exclusive of the quantity consumed in the banked fires.

The coal and its refuse were weighed upon scales that had been tested by a sealer of weights; and to guard against error the coal was weighed in parcels of 200 pounds each. The refuse was weighed dry.

In commencing an experiment the banked fire was thoroughly cleaned of ashes and clinker, which reduced it to just enough for kindling when spread out over part of the grate and covered with fresh coal. This quantity—quite a small one—was estimated by the eye, and at the close of the experiment the coal

was so nearly burned out that it only furnished, when the fires were cleaned and banked, the same amount as was used to kindle from. To maintain the steam pressure during thirteen hours of rest required about 1800 pounds of coal to be gradually expended. As this was the regular method of working, the firemen had become so practised that they could manage their fires to leave, within a few pounds, the same weight of coal upon the grates at the close of an experiment that was upon them at the beginning.

The boiler and engine, though in the same building, occupied separate apartments, and during the experiments there was a free circulation of air through both.

The engine, pumps, and boilers were in ordinary working order, and under the actual conditions of the experiments performed well. The pumps had, according to some experiments made previously for the Water Commissioners, a loss of action of about 2 per centum, ascertained by a comparison of their calculated discharge with their real discharge as measured in the reservoir.

The pressure required to work the engine and pumps *per se* was 0·984 pound per square inch of piston, and cannot be considered small when the large area of the piston is taken into view, and the absence of shafting.

The heating surface in the boilers was covered with a hard scale of about $\frac{1}{32}$ of an inch in thickness, derived from the lime in the feed-water. As the boilers had been in regular operation but 1107¾ hours, extending through about 4½ months, and had been "blown out" twice a week during that time, carrying steam, too, of only 13 pounds pressure, this formation must be regarded as excessive, and evidencing a large proportion of lime in the water. The construction of the boiler is such as to render access to the lower flues impossible, so that, notwithstanding partial cleanings, scale must continue to accumulate, and, as a consequence, steadily reduce the evaporative economy of the boiler.

The coal used was a very good quality of hard anthracite from the Lackawanna district in Pennsylvania; it gave for an average only $11\frac{8}{10}$ per centum refuse, three-fourths of which were clinker. The combustion, without being forced, was about as rapid as the draught could command. There was no excess of boiler power, and the full capacity was required to get steam enough for the pumping done.

When the boilers were erected it was supposed that two of them would furnish the requisite supply of steam, and that the third would remain as a spare boiler for cleaning, repairs, &c., from which it appears that the anticipated consumption of coal was only two-thirds of the actual consumption. This nearly corresponds to the difference between the anticipated duty of 60,000,000 pounds of water raised one foot high by 100 pounds of coal, and the duty actually realized of a little over 42,000,000 pounds.

It must be particularly noted that when the jacket was used, it was filled with steam of the boiler pressure, which was 5½ pounds per square inch greater than the initial pressure in the cylinder, and of a correspondingly higher temperature.

## EXPLANATION OF THE TABLE CONTAINING THE DATA AND RESULTS OF THE EXPERIMENTS.

The subjoined Table contains the Observed Data and the Calculated Results of the Experiments, grouped to facilitate their comprehension and to make reference easy.

TOTAL QUANTITIES. Line 1 contains the aggregate duration of the experiments. The experiments with steam admitted to the jacket were made during eleven consecutive hours of July 6th; nine consecutive hours of July 7th, eleven consecutive hours of July 9th, and eleven consecutive hours of July 13th, making an aggregate of forty-two hours. Circumstances beyond my control regulated this distribution; otherwise I should have continued the experiment uninterruptedly during the whole time. For the same reasons, the experiments with steam excluded from the jacket, were made during eleven consecutive hours of July 10th, 11th, and 12th, and ten consecutive hours of July 14th,—making an aggregate of forty-three hours. During the whole of the experiments the weather was fine and clear; and each day's trial commenced at 7 o'clock in the morning. The following quantities in the table are the totals and means of the whole time.

The number of double strokes made by the piston (line 2) was recorded by a counter worked by the engine. A double stroke is a movement of the piston twice through the cylinder. The average length of a stroke of the steam-piston was, by actual and often repeated measurement, only 9¾ feet, although intended for 10 feet. The length of the stroke of the pump pistons was likewise 9¾ feet,—the same as that of the steam-piston. The pumps being single-acting and two in number—one at each end of the beam,—the water delivery per double stroke of steam-piston, was the displacement of the pistons of both pumps less regurgitation through valves, leakage of pistons, and exclusion of water by air.

The total weight of water pumped, as given on line 3, is the product of the aggregate displacement in cubic feet of both pump pistons per stroke of 9¾ feet, by the total number of double strokes made (line 2), by the weight (62·231 pounds) of a cubic foot of water at 74° Fahr., and by the height of the lift, 160 feet. This method of computation is adopted to obtain a quantity for the "duty" (line 40) comparable with the "duty" given for other pumping engines calculated in the same manner. The result is purely comparative, according to a conventional standard.

The weight of feed-water (line 4) pumped from the tank into the boiler, is the product of the cubic feet of water by actual measurement, pumped out of the tank, by its weight (61·918 pounds) per cubic foot at the experimental temperature (line 21) of 105° Fahr.

The total weight of steam condensed in the cylinder-jacket, (line 5) was obtained by continuously drawing off the water from the jacket, as fast as formed into a tank, by means of a small pipe and cock. The number of cubic feet, by actual measurement, filled of this tank, multiplied by the weight of the water at the temperature when measured, gives the quantity on line 5.

All the coal used (line 6) was carefully weighed in parcels of 200 pounds each, and the refuse withdrawn from the furnaces was weighed with equal care. This refuse is the whole quantity withdrawn, there being no allowance made for any unburnt fragments of coal among it; in fact there appeared none worth estimating, for the anthracite being very hard and composed entirely of lumps, and the fires being but little stirred and allowed to thoroughly burn down, nothing except earthy matter escaped combustion.

The quantity of combustible consumed (line 8), is the weight of coal (line 6), minus the weight of refuse (line 7). On line 9 is the per centum that the quantity on line 7 is of the quantity on line 6.

QUANTITIES PER HOUR. As the experiments were of unequal duration in the two cases of using steam in the jacket, and of excluding it from the jacket, the quantities on lines 4, 5, 6, and 8, are given per unit of time on lines 10, 11, 12, and 13; while lines 14 and 15 contain the weights of coal and combustible burned per unit of time per square foot of grate surface.

ENGINE. Line 16 contains the number of double strokes per minute made by the engine piston. The length of its single stroke was 9¾ feet.

Line 17 contains the point at which the steam was cut off in the cylinder from the commencement of the stroke of the piston. It is the mean, by actual measurement of all the indicator diagrams taken. The steam followed the piston six-tenths of the stroke, and expanded through four-tenths only.

Line 18 gives the pressure in the condenser below the atmosphere by ALLEN'S spring gauge, graduated to denote inches of mercury.

Line 19 gives the height of the barometer in inches of mercury.

TEMPERATURES. On line 20 is the temperature of the water pumped: it was taken in the pump-well.

On line 21 is the temperature of the feed-water when in the tank. Line 22 contains the temperature of the same feed-water when entering the boiler after having passed through the heater in the boiler flue. The feed-water was drawn from the air-pump hot-well into the tank and thence fed to the boiler through this heater.

Line 23 contains the temperature of the engine room in which there was a free circulation of air.

STEAM PRESSURES. Line 24 contains the mean boiler pressure as given by an ALLEN spring gauge attached direct to the shell of each boiler.

Lines 25, 26, 27, 28, and 29, contain the cylinder pressures as given by the indicator, and are the means of all the diagrams taken. These pressures are all counted from the zero line or line of no pressure, which is taken, according to the barometer (line 19), to have been during the experiments with steam admitted to the jacket, 14·35 pounds per square inch below the then pressure of the atmosphere; and during the experiments with steam excluded from the jacket, 14·38 pounds. As the spring of the indicator is graduated to pounds per square inch, if the number of these pounds below or above the atmospheric line, be subtracted from or added to the atmospheric pressures of 14·35 and 14·38 pounds per square inch, there will be obtained the pressures above the zero or line of no pressure in the particular cases of the experiments.

Line 25 contains the pressure at the commencement of the stroke of the piston, whence it gradually falls by throttling until, at the point of cutting off, it has the pressure contained in line 26. Line 27 contains the pressure at the end of the stroke of the piston just before the opening of the exhaust-valve. Line 28 contains the back pressure against the piston. Line 29 contains the mean gross effective pressure on the piston; it is the average of the ordinates of all the indicator diagrams taken without deduction for the friction of the engine.

POWER. Line 30 contains the gross effective indicated horses power developed by the engine. It is calculated from the speed of piston (line 16) and from the pressure on the piston (line 29). Area of piston 6303·191 square inches, stroke of piston 9¾ feet.

Line 31 contains the total power developed by the engine. It is calculated from the speed of the piston (line 16) and the sum of the pressures on lines 28 and 29, namely, 14·7 pounds per square inch of piston.

Line 32 contains the net indicated horses power applied to pumping water. It is calculated from the area, 964·42 square inches, of the pump-piston, the pressure of 72 pounds per square inch of pump piston per indicator, and the same speed of piston as for the engine powers on line 30 and 31.

On line 33 is the per centum which the power on line 32 is of the power on line 30.

Line 34 contains the remainder of the per centum on line 33 deducted from 100·0, and is the per centum of the gross horses power (line 30) which is absorbed in working the engine and pumps *per se*. Otherwise expressed, this power to work the engine and pumps *per se* is 0·984 pound per square inch of the steam piston.

The quantities on line 35 are obtained by dividing those on line 12 by those on line 30. The quantities on line 36 are obtained by dividing those on line 12 by those on line 31. The quantities on line 37 are obtained by dividing those on line 12 by those on line 32. The quantities on line 38 are obtained by dividing those on line 13 by those on line 30. The quantities on line 39 are obtained by dividing those on line 13 by those on line 31. The quantities on line 40 are obtained by dividing those on line 13 by those on line 32. The quantities on lines 41, 42, and 43 are obtained by dividing, respectively, that on line 10 by those on lines 30, 31, and 32.

Duty. The quantities contained on line 44 are obtained by dividing those on line 3 by those on line 6.

Total Evaporation. Line 45 contains the total weight of steam discharged from the cylinder into the condenser. It is calculated from the displacement in cubic feet of the steam-piston per stroke, plus the cubic feet of space comprised between the piston at the end of its stroke and the exhaust-valve, less the value of the back pressure (440·585 cubic feet), from the number of strokes of piston made, and from the weight (0·026295 pound) of a cubic foot of steam of the pressure at the end of the stroke of the piston, as given by the indicator and barometer and contained in line 27. The weight of the cubic foot of steam is from Fairbairn's formulæ.

Line 46 contains the weight of steam condensed in the cylinder to furnish the heat transmuted into the total power developed by the engine. It is calculated from Joule's thermal equivalent of one pound of water raised 1° on Fahrenheit's scale for every 772 foot-pounds developed by the engine.

Line 47 contains the same quantity as line 5. It is the weight of steam condensed in the jacket of the cylinder by external radiation and in sustaining the temperature of the metal of the cylinder, &c. It was obtained by direct measurement corrected for temperature.

Line 48 contains the weight of steam consumed by the small auxiliary engine employed to feed the boilers. This engine worked without expansion, and the steam pressure in its cylinder was ascertained approximately by a loaded valve. The whole quantity is so trifling that its entire omission would not have practically affected the results of the experiments, particularly as it is nearly the same in both cases.

Line 49 contains the sum of the quantities on lines 45, 46, 47, and 48, corrected for the difference in evaporative effect between evaporating from the actual temperature (112° Fahr.) of the feed-water, and from the same feed-water at the temperature of 100° Fahr. This quantity is given to render the results com parable with those of other experiments reduced to the same temperature.

Line 50 contains the weight of feed-water used from the tank and given in line 4, corrected in the same manner for the difference between the temperatures of 112° and 100° Fahr. of feed-water.

Line 51 contains the sum of the quantities on lines 45, 46, 47, and 48, corrected in the same manner and for the same purpose as the quantities on line 49, but for the difference in evaporative effect between evaporating from the actual temperature (112° Fahr.) of the feed-water, and from the same feed-water at the temperature of 212° Fahr., instead of 100°. This reduction of the results to the two temperatures of 100° and 212° Fahr. is for the convenience of the reader, as sometimes the one and sometimes the other is employed for the standard; but, if possible, that temperature should always be used which approximates the closest to what was the experimental temperature of the feed-water; because the quantity of heat required for raising the temperature of a given weight of water one degree varies slightly with the temperature of the water. But in the case of evaporation this fact will not sensibly affect the result; for in evaporating, say from 212° Fahr., the latent heat will be 966·6° and the sensible heat 212°, making a total of 1178·6°, and it is of this large figure that the slight difference of heat required to raise the temperature of the feed-water through the same number of degrees from different temperatures will be the fraction expressing the error.

Line 52 contains the weight of feed-water used from the tank, corrected, as before, for the difference between the temperatures 112° and 212° Fahr. of the feed-water.

Economic Evaporation. From line 53 to line 60, both inclusive, is contained the weight of water evaporated from temperatures 100° and 212° Fahr. by one pound of coal and by one pound of combustible. The results are given both for the tank and for the indicator, including in the latter the weight of steam condensed in the cylinder-jacket. They show the discrepancies, under the different conditions of the experiments, of the two methods of measurement.

The quantities on lines 53, 55, 57, and 59 are the quotients of the sum of the quantities on lines 45, 46, 47, and 48, corrected for difference of temperature and divided respectively by the quantities on lines 6 and 8. The quantities on lines 54, 56, 58, and 60 are the quotients of the quantities on lines 50 and 52, divided by the quantities on lines 6 and 8.

Condensation. The quantity on line 61 is the per centum which the quantity on line 46 is of the quantity on line 4. The quantity on line 62 is the per centum which the difference of the quantities on lines 49 and 50 is of the quantity on line 49. It does not, however, express the condensation due to all other causes than the production of the power, because it is exclusive of that part of the condensation during the first portion of the stroke of the piston, which has been re-evaporated under the lessening pressure during the last portion.

TABLE CONTAINING THE DATA AND RESULTS OF THE EXPERIMENTS MADE WITH THE PUMPING ENGINE OF THE BROOKLYN WATER WORKS, JULY, 1860.

| NUMBER OF LINE. | | | STEAM ADMITTED TO JACKET. | STEAM EXCLUDED FROM JACKET. |
|---|---|---|---|---|
| 1 | TOTAL QUANTITIES. | Duration of the Experiment; in hours, | 42 | 43 |
| 2 | | Number of Double Strokes made by the Piston; per Counter, | 20,878 | 20,822 |
| 3 | | Pounds of Water elevated 160 feet, | 26,498,980,741 | 26,426,159,908 |
| 4 | | " of Feed-water pumped from the Tank into the Boilers, | 537,178 | 550,497 |
| 5 | | " of Steam condensed in Cylinder-jacket, | 9,164 | . . . |
| 6 | | " of Lackawanna Anthracite consumed, | 62,880 | 65,420 |
| 7 | | " of Refuse withdrawn from furnaces, | 6,974 | 7,523 |
| 8 | | " of Combustible consumed, | 55,856 | 57,897 |
| 9 | | Per centum of the Refuse from the Anthracite, | 11·1 | 11·5 |
| 10 | QUANTITIES PER HOUR. | Pounds of Feed-water pumped from the Tank into the Boilers, | 12,789·95 | 12,802·26 |
| 11 | | " of Steam condensed in Cylinder-jacket, | 218·18 | . . . |
| 12 | | " of Lackawanna Anthracite consumed, | 1,496 | 1,522 |
| 13 | | " of Combustible consumed, | 1,330 | 1,346 |
| 14 | | " of Anthracite consumed per square foot of grate, | 13·30 | 13·53 |
| 15 | | " of Combustible " " " " | 11·82 | 11·96 |
| 16 | ENGINE. | Number of Double Strokes per minute made by the engine-piston, | 8·087 | 7.877 |
| 17 | | Steam Cut off at, from commencement of stroke of piston; per Indicator Diagrams, | 0·60 | 0·60 |
| 18 | | Vacuum in Condenser; in inches of mercury, per Allen's spring gauge, | 26¼ | 26¼ |
| 19 | | Barometer, | 29·28 | 29·34 |
| 20 | TEMPERATURES IN DEG. FAHR. | Of the Water pumped, | 74 | 74 |
| 21 | | " Feed-water in the Tank, | 105 | 105 |
| 22 | | " " entering the Boilers, | 112 | 112 |
| 23 | | " Engine Room, | 76 | 77 |
| 24 | STEAM PRESSURES. | In pounds per square inch above atmosphere, in Boilers; per Allen's spring gauge, | 12·0 | 12·0 |
| 25 | IN CYLINDER PER INDICATOR. | In pounds per square inch above zero; at beginning of stroke of piston, | 20·9 | 20·9 |
| 26 | | " " " " at point of cutting off the steam, | 15·4 | 15·4 |
| 27 | | " " " " at end of stroke of piston, | 10·0 | 10·0 |
| 28 | | " " " " against the piston during its stroke, | 2·7 | 2·7 |
| 29 | | Mean gross effective pressure in pounds per square inch on piston during its stroke, | 12·0 | 12·0 |

(TABLE CONTINUED.)

| NUMBER OF LINE. | | | | STEAM ADMITTED TO JACKET. | STEAM EXCLUDED FROM JACKET. |
|---|---|---|---|---|---|
| 30 | POWER. | ABSOLUTE. | Gross Effective Horses Power developed by the Engine; per Indicator, | 361·45 | 352·06 |
| 31 | | | Total Horses Power developed by the Engine; per Indicator, . | 442·78 | 431·27 |
| 32 | | | Net Horses Power applied to the pumping of water; per Indicator, | 331·82 | 323·21 |
| 33 | | | Per centum of Gross Effective Horses Power utilized, . . | 91·8 | 91·8 |
| 34 | | | Per centum of Gross Effective Horses Power expended in working the engine and pumps, *per se*, . . . . . . | 8·2 | 8·2 |
| 35 | | ECONOMIC. | Pounds of Coal consumed per hour per Gross Effective Indicated Horse Power, . . . . | 4·189 | 4·323 |
| 36 | | | Pounds of Coal consumed per hour per Total Indicated Horse Power, . . | 3·379 | 3·529 |
| 37 | | | Pounds of Coal consumed per hour per Net Indicated Horse Power applied to pumping, . . . . | 4·508 | 4·709 |
| 38 | | | Pounds of Combustible consumed per hour per Gross Effective Indicated Horse Power, . . | 3·680 | 3·823 |
| 39 | | | Pounds of Combustible consumed per hour per Total Indicated Horse Power, . . . . | 3·004 | 3·121 |
| 40 | | | Pounds of Combustible consumed per hour per Net Indicator Horse Power applied to pumping, . . | 4·008 | 4·164 |
| 41 | | | Pounds of Feed-water consumed per hour per Gross Effective Indicated Horse Power, . . . . | 35·885 | 36·364 |
| 42 | | | Pounds of Feed-water consumed per hour per Total Indicated Horse Power, . . | 28·886 | 29·685 |
| 43 | | | Pounds of Feed-water consumed per hour per Net Indicated Horse Power applied to pumping, . . . . | 38·545 | 39·610 |
| 44 | DUTY. | | Pounds of water pumped one foot high per pound of coal, . . | 421,757 | 403,946 |
| 45 | TOTAL EVAPORATION | FROM FEED-WATER. | Pounds of Steam discharged from Cylinder into Condenser; calculated from the pressure of the Steam at end of stroke of Piston, . . | 472,165·701 | 470,868·161 |
| 46 | | | Pounds of Steam equivalent to the Heat annihilated in the Cylinder to produce the total power of the engine; calculated from JOULE's equivalent, . . . . . . . . | 44,327·076 | 44,203·505 |
| 47 | | | Pounds of Steam condensed in Cylinder-jacket, . . | 9,164·000 | . . . |
| 48 | | | Pounds of Steam consumed in working auxiliary feed-pump, . | 900·000 | 922·000 |
| 49 | | | Total number of pounds of Steam evaporated from water at 100° Fahr., per INDICATOR, and condensed in Cylinder-jacket, . . | 520,749·165 | 510,302·559 |
| 50 | | | Total number of Pounds of Steam evaporated from water at 100° Fahr., per TANK, . . . . . . . | 531,253·242 | 544,425·342 |
| 51 | | | Total number of Pounds of Steam evaporated from water at 212° Fahr., per INDICATOR, and condensed in Cylinder-jacket, . . | 580,507·266 | 568,861·869 |
| 52 | | | Total number of Pounds of Steam evaporated from water at 212° Fahr., per TANK, . . . . . . . | 592,216·730 | 606,900·381 |

(TABLE CONTINUED.)

| NUMBER OF LINE. | | | STEAM ADMITTED TO JACKET. | STEAM EXCLUDED FROM JACKET. |
|---|---|---|---|---|
| 53 | ECONOMIC EVAPORATION. | Pounds of Steam evaporated from water at 100° Fahr. by one pound of Coal; by INDICATOR, | 8·288 | 7·800 |
| 54 | | Pounds of Steam evaporated from water at 100° Fahr. by one pound of Coal; by TANK, | 8·455 | 8·322 |
| 55 | | Pounds of Steam evaporated from water at 212° Fahr. by one pound of Coal; by INDICATOR, | 9·239 | 8·695 |
| 56 | | Pounds of Steam evaporated from water at 212° Fahr. by one pound of Coal; by TANK, | 9·425 | 9·277 |
| 57 | | Pounds of Steam evaporated from water at 100° Fahr. by one pound of Combustible; by INDICATOR, | 9·323 | 8·814 |
| 58 | | Pounds of Steam evaporated from water at 100° Fahr. by one pound of Combustible; by TANK, | 9·511 | 9·403 |
| 59 | | Pounds of Steam evaporated from water at 212° Fahr. by one pound of Combustible; by INDICATOR, | 10·393 | 9·825 |
| 60 | | Pounds of Steam evaporated from water at 212° Fahr. by one pound of Combustible; by TANK, | 10·603 | 10·482 |
| 61 | CONDENSATION. | Per centum of the Steam generated in the Boiler, condensed to produce the total power of the engine according to JOULE'S equivalent, | 8·25 | 8·03 |
| 62 | | Per centum of the Steam generated in the Boiler, not accounted for by the Indicator including the weight of Steam condensed in the Jacket, | 1·98 | 6·27 |

## REMARKS ON THE RESULTS OF THE EXPERIMENTS.

BOILERS. The average rate of combustion was 13·41 pounds of anthracite per square foot of grate per hour, and was about equal to that of our marine boilers burning the same kind of coal with natural draught. Yet with 40·16 square feet of heating surface to one of grate, and with the products of combustion several times broken up and changed in direction during the long travel of 71½ feet, their temperature when leaving the boiler was 500° Fahr.

The evaporation per pound of fuel also evidenced a mediocre economy; for on an average the pound of coal evaporated by tank measurement only $\left(\frac{8\cdot455+8\cdot322}{2}=\right)$ 8·388 pounds of water from the temperature of 100° Fahr., and the pound of combustible only $\left(\frac{9\cdot511+9\cdot403}{2}=\right)$ 9·457 pounds of water.

A portion of this inferior result may be referred to the insufficient cross area of the chimney in proportion to the rapidity of the combustion. This area is but a trifle more than one-ninth the area of the grate-surface, and but a little more than one-half the area of the upper flues. It is doubtful whether so small a proportion of section of chimney could pass air enough for a good economic combustion when burning 13·41 pounds of anthracite per square foot of grate per hour; and that it was too small is shown by the fact that with a chimney 100 feet high at a temperature of 500° Fahr., the combustion could not be

made to much exceed that weight. Again, the fact that fully three-fourths of the refuse was hard, vitrified clinker, proved an unusually high temperature in the furnaces, a sure result of an insufficient supply of air, notwithstanding the holes through the furnace doors. Apart, however, from the low evaporation, which may have resulted from imperfect combustion in generating the heat, there remains the fact that the heating surfaces did not properly absorb the heat thrown upon them. To pass the gases into the chimney at 500° Fahr. under the conditions, proves the surfaces to transmit the heat badly. An examination of the design of the boiler shows that it must have a very deficient circulation. The gases at their highest temperature are first applied to the surface of the water, and then, as their temperature is reduced, they descend and are applied successively to lower strata of water. The top strata is, therefore, much hotter than the lower ones, and the temperature probably decreases gradually from the top to the bottom, which would, by preventing circulation, produce complete stagnation. In this state the heating surfaces can only be freed of the steam-bubbles generated upon them by the ascensional power of the bubbles, and to obtain sufficient for this purpose the bubbles must have a size that will enable their buoyancy to overcome their adhesion to the iron surface, and the cohesion of the superincumbent weight of water. The accumulation of this size is a work of time, and while the steam-bubbles remain in contact with the metal they exclude the water. If, on the contrary, there be a violent circulation of the water, the mechanical action of the currents will sweep off the bubbles as fast as formed, and thus enable the same evaporation to be obtained from a greatly less surface than in the case of boiling off stagnant water with the highest temperature applied at the uppermost strata.

The slow absorption of heat in this boiler, due to the want of circulating currents, places it really low in the scale of economy, considering its large proportion of heating to grate-surface, and the number of break-ups and changes in the direction of its gases, and also the low temperature of the steam carried. It has been generally supposed that this type of boiler gave the greatest economical result of any of the flue kind, and the belief has been mainly founded on the idea that as the gases at their highest temperature were first applied to the hottest part of the water, and then at reduced temperatures to proportionally cooler parts of the water, their heat would be more completely extracted than in the case of boilers where the reverse conditions obtain. However, here, as in many other things, the theoretic deduction was made without estimating the effect of other and antagonistic causes acting at the same time and with greater effect. My own experience has been that, *ceteris paribus*, the highest evaporative economy was found in boilers having the most rapid circulation, though obtained by applying the gases at their greatest temperature to the bottom instead of to the top strata of the water.

EFFECT OF THE STEAM-JACKET. If, in the experiments made with "Steam admitted to the Jacket," we compare the sum of the weight of steam discharged from the cylinder at the end of the stroke of its piston (line 45) and of the weight of steam consumed in working the auxiliary feed-pump (line 48), with the total weight of feed-water pumped into the boiler (line 4) less the weight of steam condensed in the jacket (line 47), we shall find a discrepancy of 54,948·299 pounds which must have been due to condensation in the cylinder, although it is not certain that it includes all of that condensation, for if any of the steam underwent condensation during the first portion of the stroke of the piston, some of the resulting water would inevitably be re-evaporated during the last portion and consequently be included in the above quantity. Of these 54,948·299 pounds, 44,327·076 pounds (line 46) were condensed to set free the heat annihilated in producing the total power of the engine, leaving 10,621·223 pounds or only $\left(\frac{10621{\cdot}223 \times 100}{537178{\cdot}000 - 9164{\cdot}000} = \right) 2{\cdot}01$ per centum of the weight of steam which entered the cylinder to be condensed by all other causes than the production of the power,

If, now, in the experiments made with "Steam excluded from the Jacket," we compare, in a similar manner, the sum of the weight of steam discharged from the cylinder at the end of the stroke of its piston (line 45), and of the weight of steam consumed in working the auxiliary feed-pump (line 48), with the total weight of feed-water pumped into the boiler (line 4) we shall find a discrepancy of 78,706·839 pounds which, as in the other case, must have been due entirely to condensation in the cylinder, although probably not including the whole of that condensation. Of these 78,706·839 pounds, 44,203·505 pounds (line 46) were condensed to produce the power, leaving 34,503·334 pounds or $\left(\frac{34503{\cdot}334 \times 100}{550497{\cdot}000} =\right)$ 6·27 per centum of the weight of steam which entered the cylinder to be condensed by all other causes than the production of the power.

From the foregoing it will be perceived that the condensation *in the cylinder* due to all other causes than the production of the power was with "Steam admitted to the Jacket," 2·01 per centum of the weight of steam entering the cylinder; and with "Steam excluded from the Jacket," 6·27 per centum. The difference between these per centums, however, is not all gain due to the use of steam in the jacket; there must be deducted from it the condensation in the jacket, which amounted to 9,164 pounds of steam (line 47). If this be included, as it should be, we shall have, when admitting steam to the jacket, a condensation in the cylinder *and jacket* due to all other causes than the production of the power, of (10,621·223 + 9,164·000 =) 19,785·223 pounds, equal to $\left(\frac{19{,}785{\cdot}223 \times 100}{537178{\cdot}000} =\right)$ 3·68 per centum of the total weight of steam evaporated in the boiler. The decrease of condensation due, therefore, to the use of steam in the jacket, amounts to (6·27 — 3·68 =) 2·59 per centum of the total weight of steam evaporated in the boiler; or, in round numbers, to four-tenths of the condensation with steam excluded from the jacket.

The weight of steam condensed in the jacket amounted to (line 47), 9164 pounds or $\left(\frac{9164 \times 100}{537178} =\right)$ 1·75 per centum of the weight of steam evaporated in the boiler. Only the sides of the cylinder were steam-jacketed; the ends were hollow and well protected by non-conductors. The area of the sides of the cylinder was 259·55 square feet, and of the two ends 88·35 square feet,—total 347·90 square feet: the jacketed portion was, therefore, $\left(\frac{259{\cdot}55 \times 100}{347{\cdot}90} =\right)$ nearly 75 per centum of the whole. If the cylinder ends had been jacketed, and the condensation in them had been *pro rata*, we should have had for a completely jacketed cylinder, a jacket condensation of 2⅓ per centum of the total weight of steam used; and it would have reduced the condensation in the cylinder, due to other causes than the production of the power, to about 2¼ per centum of the weight of steam evaporated in the boiler.

The temperature of the steam in the jacket was 244° Fahr.; the temperature of the initial steam in the cylinder was 230° Fahr.: and the average temperature of the steam in the cylinder during the stroke of the piston was 212° Fahr.; there was, therefore, quite an excess of temperature in the jacket over the temperature in the cylinder during every portion of the stroke of the piston.

To determine the economic gain due to the steam-jacket, we will take, in the two cases of admitting the steam to, and of excluding it from the jacket, the weight of coal and of combustible consumed per hour, and the weight of feed-water pumped into the boiler per hour, for the measures of the cost; and for the measure of the effect produced, we will take the gross effective indicated horses power developed by the engine. Of the three distinct measures of the cost, that of the weight of feed-water is by far the most exact, as it eliminates all the differences in the conditions of generating the steam, such as skill in firing,

quality of fuel, &c. The next in order of exactness, is the weight of combustible consumed, as that eliminates the difference in the per centum of refuse from the coal.

In the case of the coal, we have, with "Steam admitted to the Jacket," the indicated horse power given by the consumption of 4·139 pounds per hour (line 35): and with "Steam excluded from the Jacket," we have the indicated horse power given by 4·323 pounds; making a gain with "Steam admitted to the Jacket" of $\left(\frac{4{\cdot}323 - 4{\cdot}139 \times 100}{4{\cdot}323} = \right)$ 4·26 per centum of the cost with "Steam excluded from the Jacket."

In the case of the combustible, we have, with "Steam admitted to the Jacket," the indicated horse power given by the consumption of 3·680 pounds per hour: (line 38) and with "Steam excluded from the Jacket," we have the indicated horse power given by 3·823 pounds; making a gain with "Steam admitted to the Jacket" of $\left(\frac{3{\cdot}823 - 3{\cdot}680 \times 100}{3{\cdot}823} = \right)$ 3·74 per centum of the cost with "Steam excluded from the Jacket."

In the case of the feed-water, we have with "Steam admitted to the Jacket" the indicated horse power given by the consumption of 35·385 pounds per hour: (line 41) and with "Steam excluded from the Jacket," we have the indicated horse power given by 36·364 pounds; making a gain with "Steam admitted to the Jacket" of $\left(\frac{36{\cdot}364 - 35{\cdot}385 \times 100}{36{\cdot}364} = \right)$ 2·69 per centum of the cost with "Steam excluded from the Jacket." This last determination I accept as the true result.

It thus appears that, even with a cylinder of the largest size the use of the steam-jacket is beneficial, and had the ends of the cylinder been jacketed as well as the sides, the gain would have risen to probably at least 4 per centum.

It must be distinctly borne in mind, that the foregoing results of the steam jacket, as well as the absolute amounts of condensation in the cylinder, apply with exactness only to a cylinder of the given dimensions and using steam of the given quantity in the given time and in the given manner; and that these results will vary with variations of cylinder dimensions, manner of using the steam and quantity used per unit of time.

To the benefit derived from the prevention of condensation in the cylinder by the steam-jacket, may be added that due to the higher temperature of the water used to supply the steam for it. For as the water resulting from the condensation of the steam in the jacket, flows back to the boiler, it has the boiler temperature, and of course requires less heat for its re-evaporation than if it had been fed in from the hot-well. In the case of the experiments the total heat of the boiler steam was 1188° Fahr. and its sensible temperature 244° Fahr. To evaporate, then, the water supplying the jacket with steam, (1188 — 244 =) 944° Fahr. would have to be imparted, whereas if this water had been taken from the same source as the other feed-water, it would have had a temperature of 112° Fahr., and there must have been imparted to it (1188 — 112 =) 1076° Fahr. instead of 944°, making a difference of $\left(\frac{1076 - 944 \times 100}{1076} = \right)$ 12·27 per centum. But as the weight of water drawn from the jacket was only 1·75 per centum of the whole weight of feed-water, the gain due to this cause would be only 12·27 per centum of 1·75 per centum or about ⅕th of one per centum of the whole weight of feed-water. This result, however, can only be produced when the bottom of the jacket is above the level of the water in the boiler, so that the water of condensation can flow by gravity from one to the other.

Duty. The water was pumped through the absolute height of 160 feet, ascertained by careful leveling from the average level of the water in the pump-well to the level of its discharge into the reservoir. The *duty* of the engine, that is to say, the commercial value of its performance, was equal to the actual weight of water measured in the reservoir, lifted through the above height by the actual weight of coal

consumed. The labor done at the pumps was of course greater than this net weight of water lifted, by the resistance of the suction and delivery pipes, and pump barrels, to the movement of the water through them; by the loss of water due to regurgitation through the valves,—to leakage past the piston,—and to the displacement of water in the pumps by air. A slight air leak on the suction side may make the value of the last item very large.

The "duty," calculated without any corrections whatever, by the displacement of the pump-piston, expressed in pounds of water, into the absolute height of lift in feet, divided by the number of pounds of coal consumed, is popularly employed for the purpose of comparing the performances of different pumping engines; but improperly so, if it be intended as a measure of the real excellence of the machinery employed. For even with machinery equally well designed and executed, in equally good order, using the same steam pressure in the cylinder, with the same measure of expansion, the duty will vary with the absolute size of engine, with the length and diameter of main or delivering pipe and its curvature, with the quality of the coal, with the evaporative efficiency of the boiler, &c., and even with the temperature of the water pumped, for this temperature exercises no small influence on the economic performance, and the warmer the water the better will be the result, as a consequence both of its lessened specific gravity and of its greater fluidity. The difference between the cohesive resistance of water at 32° and at 75° Fahr, is very marked. The duty, therefore, as usually calculated, is not strictly a measure of anything; it is almost as conventional as nominal horse power;—it does not give the weight of water actually delivered into the reservoir, because deductions must be made for loss of action at the pumps, and for differences of temperature. The loss of action is known in different cases to have been from $\frac{1}{30}$th to $\frac{1}{8}$th of the calculated quantity without having been suspected. To make a proper comparison between two pumping engines, each condition must be appreciated separately, and proper credit given where it belongs. A good boiler and poor engine in one case, may give as good an aggregate result as a poor boiler and a good engine in another; and a difference of 20 per centum in the quality of the fuel (often the case) is enough to overbalance great inequality of engineering excellence.

The determination of the friction of the engine and pumps *per se* can be ascertained by excluding the water from the latter, and taking an indicator diagram from the steam-cylinder. Different types of engines of the same dimensions of cylinder will have different frictions. The propriety of the proportions of the organs of the engine can be determined from a diagram taken when doing the regular work. This will show whether the steam and exhaust-valves are of the proper dimensions for letting the steam in and out of the cylinder, whether they are adjusted properly, and whether the vacuum in the cylinder is of the proper excellence, &c. All these things can be made equally good with all types and sizes of engines. The resistance of the suction and delivery pipes, and of the pump-barrels and valves, can be ascertained by indicator diagrams taken from the cylinder and pump. The statical pressure on the pump-piston is known from the absolute height of the lift. Deducting the friction of the engine and pumps *per se* from the cylinder diagram, the remaining pressure on the steam-piston, less the statical water pressure on the pump-piston, will be the pressure absorbed by the resistance of the water to movement in the suction and delivery pipes, pump-barrels, and valves. Or, more directly, the pressure given by a diagram taken from the pump, plus the pressure due to the column of water below the level of the indicator, and less the statical water pressure due to the absolute lift, will give the resistance of the suction and delivery pipes, &c.

The quantity of feed-water consumed, measured in a tank previously to being pumped into the boiler—care being taken that there is neither leakage nor priming,—compared with the total indicated horse power developed by the engine, will accurately express the comparative excellence of different methods of

using steam, provided the same dimensions of cylinder be employed. The effects of different measures of expansion for the steam, of employing different absolute pressures, of steam-jacketing, and of superheating, can thus be appreciated with precision, all the foreign and disturbing elements being eliminated. The same tank measurement permits the evaporative efficiency of the boiler to be determined also, while the evaporative results according to the indicator diagrams from the steam-cylinder, compared with this tank measurement, show the difference of the methods of measurement.

Until accurate experiments, conducted in the manner above described and furnishing all the elements for an ultimate analysis, be made on pumping engines of different types, but of the same dimensions of cylinder and pump, working with the same measure of expansion, with the same absolute steam pressures, and with equal advantages of jacketing and superheating, &c.; in fact, with equality of everything that can be made equal, leaving the purely inherent differences due rigorously to type to determine alone the differences of result, it is impossible to pronounce as to which is the best type, and how much the best, and whether the Cornish engine, or the double-acting crank engine, or a combination of both, be preferable for pumping purposes.

In the case of the Brooklyn engine, the water was pumped, as before stated, through the absolute height of 160 feet: the statical pressure of a column of water of this height at the experimental temperature of 74° Fahr. is 69·3445 pounds per square inch. An indicator diagram taken from the top of the lower pump gave a mean pressure of $67\frac{3}{4}$ pounds per square inch. The average level of the water in the pump-well was $9\frac{3}{4}$ feet below the level of the indicator, adding the pressure due to which height to the above we have for the total dynamic pressure on the pump-pistons ($67\frac{3}{4}+4\frac{1}{4}=$) 72 pounds per square inch. The difference of ($72{\cdot}0000-69{\cdot}3445=$) 2·6555 pounds per square inch, between the statical and dynamical pressures on the pump-pistons, is what is due to the resistance offered by the pumps, their connecting pipe, and the main, to the movement of the water through them. This difference would support a column of water of the experimental temperature $6\frac{1}{8}$ feet high, adding which to the absolute height of 160 feet, we have $166\frac{1}{8}$ feet for the height of an equivalent column of water at 74° Fahr., balancing the labor of the pumps.

In contracts for several pumping engines constructed in the United States, it has been stipulated that the "duty" should be determined in the following manner, namely:—First. The friction and other resistances of the water to movement in the pipes is ascertained by pressure gauges and added to the statical pressure on the pump-piston due to the height of the column of water from the level in the pump-well to the reservoir. Second. The load in pounds on the pump-piston thus ascertained, multiplied by the length in feet of the stroke actually made by that piston, and by the number of strokes, and divided by the number of pounds of coal consumed, equals the duty. Let us apply this method to the Brooklyn engine.

We have seen in the experiments when using the steam-jacket, that—adding the value of the friction resistances, &c., of the water in the pipes, to the statical water pressure—the pressure on the pump-piston was 72 pounds per square inch; then, the stroke of the pump-piston being $9\frac{3}{4}$ feet and its area 964·42 square inches, exclusive of rod, the number of double strokes made being 20,378, and the weight of coal consumed 62,830 pounds,—the duty, according to the above method of calculation, was

$$\left(\frac{72\times9{\cdot}75\times964{\cdot}42\times20378\times2}{62830}=\right) 439{,}165 \text{ pounds raised one foot high by one pound of coal.}$$

## AVERAGE PERFORMANCE OF THE BROOKLYN ENGINE.

As it will be satisfactory to known how near the foregoing quantities, obtained during the short time of the experiments, agree with the average of the performance for a long period, I have extracted from the books of the Water Commissioners the following items which embrace the whole amount of pumping done since the engine came into their hands.

The average steam pressure in the boilers was 13 pounds per square inch above the atmosphere. The point of cutting off the steam was six-tenths of the stroke of the piston from the commencement. The vacuum in the condenser was 26½ inches of mercury. The indicator diagrams were precisely the same as those taken during the experiments, but the average length of the stroke of the piston was a little less. The coal was carefully weighed, and the amount given is that which was actually consumed in pumping.

The reader will observe that the quantities each day varied considerably, and will be able to appreciate the force of the remark that reliable results, particularly as regards the consumption of large quantities of coal, can only be obtained from the mean of many experiments. The greatest variations in the weight of coal will be found after the intervention of one or several days repose.

Banked fires and steam were kept in the boilers during the portion of each day that the engine was not pumping: the consumption of coal for this purpose averages about 1800 pounds per day, and is additional to the quantities in the subjoined table.

The following is a general summary of the contents of this table, and gives the average performance for the whole time of pumping:—

| | |
|---|---|
| Total number of hours employed in pumping, . . . . | 1,107¾ |
| " double strokes made by the engine piston, . . | 545,431 |
| " pounds of Lackawanna anthracite consumed, . . | 1,574,030 |
| Number of double strokes made by the engine piston per minute, . . | 8·206 |
| " pounds of Lackawanna anthracite consumed per hour, . . | 1421· |

In the comparable column of the foregoing table, namely, the one giving the results with "Steam admitted to the Jacket;" the average number of double strokes made by the engine-piston per minute was 8·087, and the average weight of coal consumed per hour 1496 pounds. Great care was exercised during the experiments in obtaining the *precise* weights, and if the less length of stroke of piston during the average performance be included in the comparison, it will be found that the average results for the whole time of pumping will closely approximate those of the forty-two hours experiment.

TABLE EXHIBITING THE PERFORMANCE OF THE BROOKLYN PUMPING ENGINE FROM THE TIME OF GOING INTO REGULAR OPERATION, JANUARY 24, 1860, UP TO JULY 5, 1860, BOTH INCLUSIVE.

| DATE. 1860. | | No. of consecutive hours. | No. of Double Strokes made by the Engine Piston. | Pounds of Lackawanna Anthracite consumed. | DATE. 1860. | | No. of consecutive hours. | No. of Double Strokes made by the Engine Piston. | Pounds of Lackawanna Anthracite consumed. |
|---|---|---|---|---|---|---|---|---|---|
| Jan. | 24, | 8½ | 4,527 | 14,000 | Feb. | 13, | 11 | 5,339 | 18,900 |
| " | 25, | 11 | 5,491 | 16,000 | " | 14, | 11 | 5,800 | 15,200 |
| " | 26, | 10½ | 5,400 | 16,000 | " | 15, | 11 | 5,641 | 15,600 |
| " | 27, | 11 | 5,481 | 16,000 | " | 16, | 11 | 5,699 | 15,000 |
| Feb. | 1, | 11 | 5,619 | 19,000 | " | 17, | 11 | 5,700 | 15,000 |
| " | 2, | 11 | 5,628 | 16,500 | " | 18, | 11 | 5,721 | 14,100 |
| " | 3, | 11 | 5,678 | 16,500 | " | 27, | 10½ | 5,200 | 20,000 |
| " | 4, | 11 | 5,678 | 15,600 | " | 28, | 10¾ | 5,281 | 16,300 |

(TABLE CONTINUED.)

| DATE. 1860. | | No. of consecutive hours. | No. of Double Strokes made by the Engine Piston. | Pounds of Lackawanna Anthracite consumed. | DATE. 1860. | | No. of consecutive hours. | No. of Double Strokes made by the Engine Piston. | Pounds of Lackawanna Anthracite consumed. |
|---|---|---|---|---|---|---|---|---|---|
| Feb. | 29, | 11 | 5,564 | 15,900 | May | 11, | 11 | 5,191 | 14,700 |
| March | 12, | 11 | 5,321 | 20,400 | " | 12, | 11 | 5,416 | 14,700 |
| " | 13, | 11 | 5,657 | 16,200 | " | 14, | 11 | 5,495 | 15,800 |
| " | 14, | 11 | 5,672 | 16,200 | " | 15, | 11 | 5,183 | 15,200 |
| " | 15, | 11 | 5,721 | 16,200 | " | 16, | 11 | 5,358 | 15,000 |
| " | 16, | 11 | 5,651 | 15,900 | " | 17, | 11 | 5,364 | 15,500 |
| " | 17, | 11 | 5,800 | 14,855 | " | 18, | 11 | 5,422 | 15,300 |
| " | 26, | 10½ | 5,224 | 20,400 | " | 19, | 11 | 5,388 | 15,200 |
| " | 27, | 11 | 5,548 | 15,800 | " | 21, | 11 | 4,991 | 15,000 |
| " | 28, | 11 | 5,307 | 15,000 | " | 22, | 11 | 5,448 | 15,200 |
| " | 29, | 11 | 5,488 | 14,700 | " | 23, | 11 | 5,436 | 14,750 |
| " | 30, | 11 | 5,371 | 14,700 | " | 24, | 11 | 5,547 | 15,200 |
| " | 31, | 11 | 5,400 | 14,700 | " | 25, | 11 | 5,460 | 14,000 |
| April | 2, | 11 | 5,408 | 15,645 | June | 1, | 11 | 4,523 | 18,700 |
| " | 3, | 11 | 5,430 | 15,300 | " | 2, | 11 | 5,371 | 14,400 |
| " | 4, | 11 | 5,450 | 14,700 | " | 4, | 11 | 5,272 | 15,000 |
| " | 5, | 11 | 5,402 | 15,000 | " | 5, | 11 | 5,370 | 14,166 |
| " | 6, | 11 | 5,477 | 15,000 | " | 6, | 11 | 5,410 | 13,900 |
| " | 7, | 11 | 5,534 | 15,300 | " | 7, | 11 | 5,242 | 14,000 |
| " | 9, | 11 | 5,510 | 16,200 | " | 8, | 11 | 5,208 | 13,900 |
| " | 10, | 11 | 5,500 | 15,600 | " | 9, | 11 | 5,259 | 14,000 |
| " | 11, | 11 | 5,464 | 15,600 | " | 11, | 11 | 5,261 | 16,100 |
| " | 12, | 11 | 5,436 | 14,442 | " | 12, | 11 | 5,406 | 14,100 |
| " | 13, | 11 | 5,464 | 15,000 | " | 13, | 11 | 5,406 | 14,300 |
| " | 14, | 11 | 5,517 | 13,887 | " | 14, | 11 | 5,391 | 14,600 |
| " | 17, | 11 | 5,228 | 19,836 | " | 15, | 11 | 5,401 | 14,900 |
| " | 18, | 11 | 5,496 | 15,300 | " | 16, | 11 | 5,182 | 14,100 |
| " | 19, | 11 | 5,337 | 15,200 | " | 18, | 11 | 5,302 | 16,100 |
| " | 20, | 11 | 5,192 | 14,700 | " | 19, | 11 | 5,340 | 14,600 |
| " | 23, | 12 | 5,878 | 20,400 | " | 20, | 11 | 5,220 | 14,700 |
| " | 24, | 11 | 5,378 | 15,800 | " | 21, | 11 | 5,337 | 15,000 |
| " | 25, | 11 | 5,388 | 15,300 | " | 22, | 11 | 5,389 | 14,900 |
| " | 26, | 11 | 5,377 | 15,135 | " | 23, | 11 | 5,313 | 14,700 |
| " | 27, | 11 | 5,432 | 15,100 | " | 25, | 11 | 5,218 | 15,900 |
| " | 28, | 11 | 5,462 | 15,600 | " | 26, | 11 | 5,365 | 15,200 |
| " | 30, | 11 | 5,400 | 19,000 | " | 27, | 11 | 5,288 | 16,200 |
| May | 1, | 11 | 5,242 | 15,000 | " | 28, | 11 | 5,368 | 16,100 |
| " | 2, | 11 | 5,442 | 15,000 | " | 29, | 11 | 5,348 | 16,100 |
| " | 3, | 11 | 5,288 | 14,500 | " | 30, | 11 | 5,419 | 15,800 |
| " | 4, | 11 | 5,366 | 14,700 | July | 2, | 11 | 5,467 | 18,200 |
| " | 8, | 11 | 5,182 | 14,820 | " | 3, | 11 | 5,509 | 16,500 |
| " | 9, | 11 | 5,352 | 14,700 | " | 5, | 11 | 5,211 | 16,100 |
| " | 10, | 11 | 5,297 | 15,300 | | | | | |

# REPORT OF EXPERIMENTS

MADE BY

THE BOARD OF U. S. NAVAL ENGINEERS,

## TO DETERMINE THE RELATIVE ECONOMY OF USING STEAM WITH DIFFERENT MEASURES OF EXPANSION.

L

# REPORT

# MADE TO THE NAVY DEPARTMENT

## BY THE BOARD OF U. S. NAVAL ENGINEERS,

CONVENED ON BOARD THE U. S. STEAMER MICHIGAN AT ERIE, PA., NOVEMBER 19, 1860, TO DETERMINE THE RELATIVE ECONOMY OF USING STEAM WITH DIFFERENT MEASURES OF EXPANSION.

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ERIE, PA., February 18, 1861.

SIR:—

IN obedience to your orders of November 10th, 1860, the undersigned proceeded to Erie, Pa., and made, with the machinery of the U. S. paddle-wheel steamer "MICHIGAN," as complete a set of experiments as was practicable with the valve-gear of its engines, for the purpose of determining the Relative Economy in Rapport of Fuel to Power, of using Steam with Different Measures of Expansion.

The undersigned also determined the evaporative efficiency of the boilers of the "MICHIGAN" with the different coals used at Erie.

The description of the machinery, of the manner of making the experiments, of obtaining the data, and of calculating the results, together with the reasons for the same, and the deductions therefrom, are respectfully submitted in the following

## REPORT.

The law of MARIOTTE, which merely expresses the fact that the volume of gaseous substances, at a constant temperature and unaffected by the greater or less proximity of its molecules, is inversely as the pressure, or, conversely, that the pressure is inversely as the volume, promises an enormous economy of fuel when practically applied to the use of steam expansively in an engine, and the reasons derived from an abstract consideration of the case, as well as from an examination of indicator diagrams taken from a steam-cylinder, are so specious and apparently so conclusive, that, up to within the last one or two years, the assumption of economy passed unchallenged by the engineering profession, and its whole ingenuity was directed to the contrivance of mechanism by which practical effect might best be given to the law. The

**economy of using steam expansively appeared so obvious that experiments to determine that fact were deemed supererogatory, and those that were made were confined to the determination of the relative excellence of one expansion-gear over another. The grand point of first determining whether, under proper conditions of the engine, there was any economy to be derived from any of them was neglected. Latterly, however, the fact of superior economy has been denied, and the results of experiments which have been made, come so strongly to the support of the negative, that it became a matter of vital importance to all interested in the use of steam to have this question settled in the only manner in which a settlement could be conclusive, namely, by a set of exact experiments made with an engine of at least medium size, and under the ordinary conditions of good practice. It is to supply this want that the experiments hereinafter detailed have been made. Their object was to ascertain, by the direct measurement of all the quantities involved, the true practical relative economy of various measures of expansion, and so determine the problem irrespective of purely theoretical considerations, the soundness of which in physics until confirmed or disproved by experiment, must always be a matter of uncertainty.**

**The selection, by the Navy Department, of the machinery of the U. S. Steamer "MICHIGAN" for making the following experiments, was determined by its appropriateness and convenience, the engines being of medium size and the vessel out of commission. The former had just been thoroughly repaired and furnished with new boilers.**

**It is proper to preface the description of these experiments with that of the machinery, embracing all the dimensions necessary to a full comprehension of it, and to enable the reader to judge for himself of the propriety and correctness of the calculations of the results from the experimental data.**

## ENGINES.

The engines are two in number, condensing, direct-acting, and inclined from the horizon at an angle of 23 degrees. They are placed side by side in the vessel with a passage-way $4\frac{1}{2}$ feet wide between them, and they are connected upon a centre shaft at right angles to each other. The engine frames are of wood. The air-pump is inclined like the cylinder, and is situated immediately beneath it, the axes of both being parallel. It is a single-acting forcing pump, with a solid piston, and one end open to the atmosphere, the receiving and delivering valves being at the bottom. The air-pump piston is actuated from a half-beam, whose fulcrum is upon one of the engine keelsons, and whose movement is derived from the cylinder crosshead through the medium of two links. The beam at half-stroke is at right angles to the axis of the cylinder. A feed-pump is placed on one side of the air-pump, and a bilge-pump on the other; they are operated by a small crosshead which derives its motion from the air-pump piston-rod. The axes of these pumps are parallel with the axis of the air-pump. The condenser is of the common jet kind, and is situated at the bottom of the air-pump and immediately beneath the cylinder. The cylinder steam and exhaust-valves are the double-poppet valve habitually employed in the United States for paddle-wheel engines. The upper and lower valve-chests are connected by a steam and an exhaust-pipe, the axes of which are parallel with the axis of the cylinder. Each end of the cylinder is provided with a relief-valve. The cylinder steam-valves are made to act as expansion-valves by means of the mechanism known as SICKELS' cut-off. As applied to these engines, the valve was tripped by its own movement, when the spring came in contact with the inclined face of a fixed cam. By this arrangement, the point of cutting off could be graduated from nearly the commencement up to $\frac{4}{9}$ths of the stroke of the piston, and from $\frac{7}{10}$ths of the stroke up to $\frac{11}{12}$ths, at which point the valve seated, without tripping, by the eccentric movement. Between

$\frac{4}{9}$ths and $\frac{7}{10}$ths of the stroke of the piston, it was impossible to cut off the admission of steam. The steam-pipe between the boiler and cross-pipe is $25\frac{1}{4}$ feet long by $17\frac{3}{8}$ inches diameter. A cross-pipe $4\frac{3}{4}$ feet long by $15\frac{1}{4}$ inches in diameter, in which the throttle-valves are placed, connects it with the two cylinders. The steam side-pipe of each cylinder is $7\frac{1}{2}$ feet long by $12\frac{1}{4}$ inches in diameter. The total length of steam-pipe, including the side-pipe of one cylinder, exposed to the refrigeration of the atmosphere is 30 feet, and as there is a slight inclination towards the cylinder, all the water resulting from the condensation of steam by the steam-pipe surface, is passed through the cylinder. The steam-pipes, side-pipes, and sides of the cylinders are protected by a thick coat of felt covered with a wooden casing. The heads of the cylinders, valve-chests, and cylinder nozzles have no covering. The lower head of each cylinder is double; the upper one is single. The exhaust valves seat when the steam-piston is 0·4666 foot from the end of its stroke. The engines occupy in the vessel a length over all of 35 feet; a breadth of 15 feet, including the $4\frac{1}{4}$ feet wide passage between them; and a height of $13\frac{1}{2}$ feet from top of keelsons to top of main pillow-blocks.

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| Diameter of the cylinder, | 36 inches. |
| Stroke of the piston, | 8 feet. |
| Diameter of the piston-rod, | $3\frac{3}{4}$ inches. |
| Space displacement of one piston per stroke, exclusive of rod, | 56·544 cubic feet. |
| Steam space comprised between the piston at the end of its stroke and the exhaust-valve, | 3·280 " |
| Diameters of the two discs composing the steam-valve, | $8\frac{1}{4}$ and $9\frac{1}{4}$ inches. |
| " " " exhaust-valve, | $8\frac{1}{4}$ and $9\frac{1}{4}$ " |
| Net area of opening through steam-valve for the induction of steam, exclusive of areas of valve-stem and valve-connexion, | 114·96 square inches. |
| Net area of opening through exhaust-valve for the eduction of steam, exclusive of areas of valve-stem and valve-connexion, | 108·38 " |
| Diameter of air-pump, | 29 inches. |
| Stroke of air-pump piston, | $31\frac{3}{8}$ " |
| Space displacement of air-pump piston per stroke, | 12 cubic feet. |
| Diameter of feed-pump and of bilge-pump, | $5\frac{7}{16}$ inches. |
| Stroke of feed-pump and of bilge-pump plungers, | $31\frac{3}{8}$ " |
| Capacity of one condenser, | 20 cubic feet. |
| " hot-well, | 27 " |
| Length of main connecting-rod between centres, | 16 feet, 5 inches. |
| Diameter of main connecting-rod in neck, | $3\frac{5}{8}$ inches. |
| " each of the journals of the fork end of the connecting-rod, | $3\frac{11}{16}$ " |
| Length " " " " | $4\frac{1}{2}$ " |
| Diameter of the crank-pin journal, | $4\frac{3}{4}$ " |
| Length " " | 7 " |
| Diameter of the main shaft journal, | $10\frac{3}{4}$ " |
| Length " " | 12 " |
| Width of the eccentric strap, | $2\frac{3}{4}$ " |
| Area of the main crosshead guide gib (2 by 9 inches), | 18 square inches, |

NOTE.—During the experiments the starboard engine alone was employed.

## PADDLE-WHEEL.

The arms, rims, and braces of the paddle-wheels are of iron; the paddles are of wood 1½ inch thick, chamfered at the edges. Each paddle is in two pieces, one on the forward and the other on the after side of the arm. Each wheel has three sets of arms, and each set consists of sixteen arms ⅜-inch by 4 inches. Each set of arms has two rims, one on each side of the paddles: the rims are ⅝-inch by 3 inches. The arms are stiffened by diagonal braces of 1½ inch diameter iron, and there are thirty-two braces to each wheel.

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| Diameter to outside of paddles, | 21½ feet. |
| Number of paddles in each wheel, | 16 |
| Width of outer half of paddle, | 14 inches. |
| " inner " | 17 " |
| Length of paddles, | 8 feet. |

NOTE.—The above is the normal surface. During the experiments this surface varied greatly, being decreased by the removal of different portions of the paddles for the different experiments. In some of the experiments, too, parts of the paddles were broken off by the ice. The immersion of the lower edge of the paddle was 32 inches.

## BOILERS.

The boilers are two in number, placed side by side, 6 inches apart, with one smoke-pipe in common at the front end. They are of the vertical water tube kind with the tubes placed above the furnaces, and were designed by the Engineer-in-Chief, SAMUEL ARCHBOLD, according to the patent of D. B. MARTIN, but with proportions differing from those adopted by the patentee, for the purpose of burning a highly gaseous coal which had not previously been done. The tubes are of drawn brass; all the other portions of the boiler are of iron. The bottoms of the shells, the ash-pits, the furnaces, and the combustion chamber, are of ⅜ths inch thick plate; the tube plates are ½-inch thick; the remaining portions of the boiler are of $\frac{5}{16}$ths inch thick plate. The tops of the furnaces are semicircular, and incline from the front towards the back at the rate of 1 to 9⅜. The tube-plates have an inclination in the same direction of 1 in 45. The water spaces are 5 inches wide. The height from the crown of the furnace to the lower tube-plate at the front of the boiler is 7 inches, and at the back of the boiler 18 inches.

The bridge walls are of cast iron, lined above the level of the grates with brick, and having, each, below the grates a perforated plate and valve for the admission and control of air to the bottom of the combustion chamber behind the bridge. The plate (one for each furnace) contains two hundred and fifty-eight holes, each $\frac{7}{16}$-inch in diameter. The valve is operated by a handle within the ash-pit, placed conveniently for the fireman. The furnace doors are 16 inches high by 18 inches wide, and semicircular on top. Each door has a register and six openings for the admission of air to the furnace; the air is distributed by a lining perforated with fifty-eight holes of ⅝-inch diameter. The area of each of the six register openings is 4⅝ square inches.

The smoke-pipe is surrounded by a steam chimney or drum 82 inches in diameter, and 30 inches in height above the top of the boiler; it is made in halves, one half for each boiler. This chimney, though riveted upon the shell, is distinct, and the steam from the boiler enters it by means of an independent pipe

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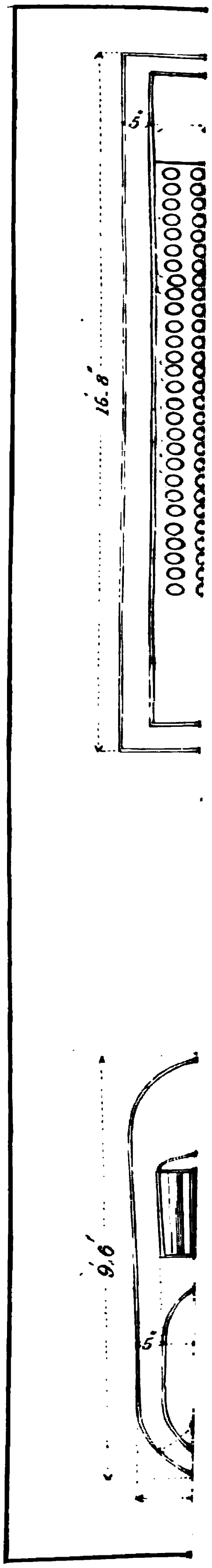

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controlled by a stop-valve. The grate-bars are in two lengths; each bar is $1\frac{1}{8}$-inch wide on the top, tapering to $\frac{7}{16}$-inch wide on bottom; it is 4 inches deep at the centre, diminishing to $2\frac{1}{2}$ inches at the ends. The air space between tops of the bars is $\frac{5}{8}$ths of an inch wide.

The tube boxes are of the same width as the furnaces, namely, 30 inches in the clear. The centres of the tubes of each row lengthwise the furnace, are not placed in the same straight line; they are slightly zigzag, being situated, alternately, in two straight lines, both starting from the centre of the extreme tube at the back of the boiler, and diverging to $\frac{1}{2}$-inch apart at the centre of the extreme tube at the front of the boiler. The tubes are secured in their plates by expanding them on one side of the plate and riveting them over on the other. The inboard furnace of each boiler has thirteen less tubes than the other two furnaces, the omission being made to preserve the full calorimeter beneath the smoke-pipe.

The peculiar features of these boilers, as distinguished from others of the same type, are:

1st. The greater length of tube-box, which is about one-half more than the patentee employs.

2d. The greater width in the clear between the tubes crosswise the furnaces, which is about two-and-a-half times that which the patentee employs.

3d. The greater calorimeter for draught between the tubes, which is about double the patentee's proportion, while the area of the smoke-pipe, instead of being equal to this calorimeter, is only about one-half of it.

4th. The employment of a much larger combustion chamber between the furnace and tubes than the patentee adopts.

5th. The furnishing of a copious supply of air, not only to the furnaces through perforations in the doors, but to the bottom of the combustion chambers, through perforations in the lower part of the bridge wall.

The purpose of these variations was to adapt the boiler to the most economical consumption of a highly gaseous coal which deposited soot in excessive quantities.

The heating surface given below is calculated for every part with which the heated gases come in contact—top, bottom, and sides—and for the external circumference of the tubes.

The whole of the boilers and steam chimney are well covered with felt.

The following are the principal dimensions of the boilers, namely:—

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| Length of each boiler at the furnaces (fore and aft the vessel), | 15 feet, 8 inches. |
| Length of each boiler on top, | 16 " 8 " |
| Breadth of each boiler, | 9 " 2 " |
| Height of each boiler, exclusive of steam chimney, | 9 " 6 " |
| Number of furnaces in each boiler | 3 |
| Width of each furnace, | 2 feet, 6 inches. |
| Length of grate-bars, | 6 " 0 " |
| Height from bottom of ash-pit to top of grate-bars at front of furnace, | 1 " 9 " |
| Height from top of grate-bars to crown of furnace " | 2 " 3 " |
| Height from top of bridge wall to crown of furnace, | 1 " 3 " |
| Total area of grate surface in both boilers, | 90 square feet. |
| Number of tubes (brass) " | 1504 |
| External diameter of the tubes, | 2 inches. |
| Internal " | 1·81 " |
| Length of the tubes, extreme, | 21 " |

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| Length of the tubes between tube-plates, | | 20 inches. |
| Distance between centres of tubes, crosswise the furnaces, | | 5 " |
| " " lengthwise " | | 3 " |
| Number of rows of tubes, crosswise each furnace, | 6 | |
| " " lengthwise " (for the full rows), | 45 | |
| Length of the space occupied by the tubes, | 11·2 feet. | |
| Aggregate absolute space between tubes, crosswise the boiler, for both boilers, | 30 square feet. | |
| Calorimeter, or area for direct draught between the tubes, for both boilers, at back end, | 30 " | |
| Calorimeter, or area for direct draught between the tubes, for both boilers, at front end, | 28½ " | |
| Heating surface in the 1504 tubes, | 1312·49 square feet. | |
| " six furnaces, | 260·03 " | |
| " six combustion chambers, | 341·37 " | |
| " six after uptakes, | 193·22 " | |
| " twelve tube plates, | 269·32 " | |
| " twelve sides of the tube boxes, | 226·66 " | |
| " two uptakes to smoke-pipe, | 86·50 " | |
| Total water heating surface in the two boilers, | | 2,689·59 " |
| Total steam heating surface in the two boilers, in the uptakes and steam-chimney, | | 84·71 " |
| Diameter of the smoke-pipe, | | 4 feet, 3 inches. |
| Height of the smoke-pipe above the grate-bars, | | 45 " |
| Steam room in the two boilers and steam chimney, | | 530 cubic feet. |
| Weight of water in boiler at 262° Fahr., measured to 4 inches above the highest point of the upper tube-plate, | | 46,450 pounds. |

## PROPORTIONS.

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| Ratio of water heating to grate surface, | 29·884 to 1·000. |
| Ratio of steam heating to grate surface, | 0·941 " |
| Ratio of grate surface to cross area between the tubes for draught, at back of boilers, | 3·000 " |
| Ratio of grate surface to cross area between the tubes for draught, at front of boilers, | 3·212 " |
| Ratio of grate surface to cross area of smoke-pipe, | 6·344 " |

## MANNER OF MAKING THE EXPERIMENTS.

The experiments were made with the starboard engine alone and both boilers. The following quantities were ascertained by direct measurement or weight:—

The number of double strokes made by the engine piston.

The number of cubic feet of feed-water pumped into the boilers, and the temperature of the same.

The number of pounds of coal consumed.

The number of pounds of refuse from the coal, in ashes, clinker, soot, and dust withdrawn from the ash-pits, furnaces, and smoke connexions of the boilers.

The vacuum in the condenser.

The steam pressure in the boilers.

The steam pressure in the cylinder throughout the stroke of the piston.

The height of the barometer.

The temperature of the injection-water, of the hot-well, of the engine room, and of the external atmosphere.

In ascertaining the above quantities the following instruments were employed:—

One of Roger's counters was connected with the engine, and registered every double stroke made by the piston.

A wooden tank, lined with sheet zinc and holding conveniently seventy cubic feet, was erected in the engine room and connected with the bilge and feed-pumps in such manner that, the feed-pump being shut-off, it could first be filled from the hot-well by the bilge-pump, and then the bilge-pump being shut off and the communication with the feed-pump opened, it could be pumped out into the boilers by the feed-pump. A hand-pump, receiving its water from outside the vessel, was also used and arranged to discharge into the tank, so that the latter could be filled each time to the precise mark. The bilge and hand-pumps delivered their water into the top of the tank through hose, which were thrown out as soon as it was filled in order to avoid the possibility of their accidentally discharging into the tank while it was being exhausted by the feed-pump. The feed-pump was connected with the tank by a copper pipe 3½ inches in diameter, fitted with a conveniently placed stop-cock. The bottom of this pipe was 1½ inch from the bottom of the tank, and the pipe itself passed over the top-edge of the tank and was bolted by a flange to the receiving orifice of the feed-pump. There being, thus, no holes in the tank for the connexion of pipes, the possibility of leakage was avoided. The bottom of the tank itself was blocked up three inches above the engine room floor, so that the whole of the tank, together with the hose, pipes, and cocks of the feed arrangement in connexion with it were in plain sight. The bilge-pump connexion with the hot-well had originally been designed for the purpose of employing the former in the event of fire, and the fire hose was now used in filling the tank. The tank could not be filled without the pumps because its top was above the level both of the hot-well discharge and of the water outboard the vessel. Previous to commencing the experiments, an iron plate was bolted over the delivery of the feed safety-valve; and all the connexions of the feed-pipe with the port engine and with the hand-pump were broken at their joints and iron plates bolted in between their flanges. Thus as every possible avenue of water leakage was perfectly guarded, and as the entire tank and feed-pipe were constantly inspected and could not have leaked without the leakage becoming immediately visible, and as the hose for filling the tank was removed as soon as it was filled, the undersigned are certain that all the water measured in the tank was delivered into the boilers, and that no more water was received into the tank than was measured. The sucking sound made by the feed-pump when the tank was exhausted, was a loud notice to close the cock in the suction-pipe and refill it; besides which, a careful and experienced man of the engineer department of the vessel, was detailed for the especial duty of attending the tank. The time required to fill the tank was too small to allow the water level in the boilers to fall more than 1½ or 2 inches, even when generating the maximum quantity of steam, between the shutting-off and the putting-on of the feed-pump. The distribution of water between the boilers was controlled by a separate screw check-valve at each boiler; one valve being opened before the other was closed: but the supply of feed-water was regulated by the cock in the suction-pipe at the tank. The temperature of the feed-water was noted by a Fahrenheit thermometer at the time when the

tank was exactly half pumped out. The dimensions of the tank in the clear were 11¼ fect by 3⅞ feet by 1¾ foot.

Every pound of the coal thrown into the furnaces was carefully weighed on one of FAIRBANKS' delicate platform scales, quite new, and which had previously been tested. The tub in which the coal was weighed was of iron, and had a capacity of 1·45 cubic foot; it was accurately counterbalanced on the scales, and filled each time to precisely the same weight, for the purpose of avoiding errors of calculation.

The refuse from the coal was all weighed on the same scales, in the same manner, and in the *dry* state.

The vacuum in the condenser was denoted by one of ALLEN'S spring gauges; it was very sensitive to the slightest variations of pressure.

The steam pressure in the boilers was denoted by an ALLEN'S spring gauge; and by an open mercurial syphon gauge. The indications of these gauges coincided.

An indicator of the New York Novelty Works' manufacture was connected by a brass pipe nine inches long, with the clearance at each end of the cylinder. This pipe was tapped into the cylinder at about its horizontal diameter. The spring of these indicators was graduated to twelve pounds per square inch. They were excellent instruments, and in all cases gave smooth, satisfactory diagrams. The pencil was made to pass three times over each diagram, and the markings coincided in all. The two indicators remained permanently attached throughout the experiments; they received their movement direct from a reducing lever worked by the air-pump crosshead.

The atmospheric pressure was given by a good Aneroid barometer borrowed from the revenue cutter of the station. It was placed in the centre of the engine room. The temperature of the engine room was noted from the thermometer attached to this barometer.

The temperature of the injection-water was noted by a thermometer immersed overboard on the side of the vessel opposite to the hot-well discharge. The temperature of the hot-well was noted by a fixed thermometer, having its bulb immersed within, and its stem protruding from, the hot-well. The temperature of the external atmosphere was noted from a thermometer suspended over the guard abaft the wheel. The thermometers were all graduated to Fahrenheit's scale.

The boilers were fitted with the usual gauge cocks and with glass water gauges. The latter enabled the height of the water within to be noted with absolute exactness by tying a piece of small twine around the glass tube. The only outlet from the boilers was through the blow-off valve. Any leakage to occur there had first to pass the blow-off valve, and next the stop-cock placed in advance of the Kingston valve. The blow-off pipe, its valves, and cocks were frequently examined during the experiments, and the undersigned are certain there was no leakage from the boilers. The boilers themselves were quite new, and double riveted; they were so tight that the water would stand in the glass gauges without appreciable fall for days. The time was noted by the engine room clock, corrected by the city time.

During the experiments the vessel was permanently covered in by a wooden housing, and remained secured to the wharf. The power of the engine was consequently consumed in paddling water aft. Each experiment lasted seventy-two consecutive hours, during which the engine was neither stopped nor slowed down, nor in any way changed in condition. In commencing an experiment the engine was operated for several hours to adjust it to the normal conditions required to be uniformly maintained during that experiment, and to bring the fires to steady action. No note was taken of the coal thus expended. When all was ready, average fires and the proper steam pressure being in the boilers, and the water in the glass gauge being at the height of four inches above the top of the upper tube plate, the level at which it was uni-

formly carried, the time and the number on the counter were noted, and the experiment began. From this time up to the end of the seventy-two hours all the quantities were systematically weighed or measured. At the end of the experiment, and precisely at the expiration of the seventy-two hours, the water in the boilers was brought to exactly the same level as at the beginning, and the fires in a thoroughly clean condition, were made the same as at the beginning, as nearly as could be estimated. With the experienced engineers and firemen employed, the error resulting from any difference in the condition of the fires at the beginning and end of an experiment on a grate surface of ninety square feet must have been small, and will be practically insignificant when distributed into the whole weight of coal consumed in the seventy-two hours. The grate surface being apportioned in six furnaces; the coal consumed being, except in one experiment, of a remarkably free burning kind, having great rapidity of ignition; the boiler power being in excess of the power developed by the engine; and the firemen being picked men with an experience of sixteen years; permitted the boiler pressure to be maintained with the variation of scarcely half a pound on the gauge during an entire experiment.

In conducting the experiments a systematic tabular record was kept, in which the following quantities were noted in appropriate columns at the end of every hour: 1st. The number on the counter; and to check any error in taking it, the double strokes made by the piston per minute during that hour were determined and placed in a parallel column. 2d. The average steam pressure in the boilers during the hour; the variations in this were very slight. 3d. The vacuum in the condenser. 4th. The height of the barometer. 5th. The temperature of the engine room, of the hot-well, of the injection-water, and of the external atmosphere. And 6th. The weight of coal thrown into the furnaces during the hour. In another column there was recorded the exact time at which each tank was filled with water, and in a parallel column the temperature of this water when the tank was half pumped out. This method of noting the tanks not only gave the exact number used, but furnished a check upon errors; for the consumption of water was so nearly uniform in its rate during an experiment that the omission of a tank would have been instantly detected by the lapse of a double time. The weight of refuse from the coal was entered in its appropriate column at the close of each watch of six hours. There was thus for each experiment seventy-two complete sets of observations arranged in columns, the footings of which gave the total quantities, or the means, as the nature of the results required.

During every hour of an experiment, two indicator diagrams were taken from the cylinder—one from each end—the time between the diagrams being half an hour. They were thus uniformly spaced over the entire experiment at half hour intervals, and were, for each experiment, one hundred and forty-four in number. Every one of these diagrams was analyzed, and its results arranged in tabular form, the atmospheric line being taken for the base. In the first column of this table was placed the initial steam pressure in the cylinder above the atmosphere—that is, the pressure at the commencement of the stroke of the piston. In the next was placed the pressure above the atmosphere at the point where the steam-valve closed, which was the pressure at the point of cutting off. In the next was placed the final pressure, or pressure at the end of the stroke of the piston, above or below the atmosphere, as the case might be. In the next was placed the mean back pressure below the atmosphere, which was the pressure of the uncondensed vapor in the cylinder resisting the piston throughout its stroke. In the next was placed the mean gross effective pressure on the piston during its stroke, which was the mean of the ordinates erected at right angles to the atmospheric line at the centres of the ten or twelve equal parts into which the diagrams were divided. And, finally, the distance from the commencement of the diagram to the point of cutting off, as shown upon it, was placed in another column. The pressures were taken carefully off from the diagrams

to the nearest tenth of a pound. The footings of all these columns divided by the number (144) of the diagrams, gave the point of cutting off and the mean cylinder steam pressures for the entire experiment, answering to the mean of the other quantities ascertained as previously described. The mean pressures thus obtained from the diagrams had now only to be converted into pressures above the absolute zero or point of no pressure, by adding them to or subtracting them from the atmospheric pressure as given by the barometer.

During all the experiments the throttle-valve was kept wide open. The boiler pressure was intentionally varied a very little, but only for the purpose of maintaining equality in the initial cylinder pressure, which, for equal boiler pressures, was slightly affected by the different speed of piston in the different experiments.

At the close of the experiments, the pressure on the piston required to operate the engine *per se* was obtained by removing all the paddles from the wheels, working the engine at various speeds ranging from 8 to 22 double strokes per minute, and taking indicator diagrams simultaneously from both ends of the cylinder, several sets for each rate of speed to obtain a reliable mean. By the differences of the pressures given by these diagrams for the different speeds of piston, there was eliminated the pressure on the piston due to the friction and resistance of the rims and arms of the wheels in the water, and the true pressure required for simply working the organs of the engine determined. This pressure was 2·1 pounds per square inch of piston, which is, of course, constant for all speeds.

During the experiments the cylinder-valves and piston were several times examined under a boiler pressure of 21 pounds per square inch, and showed no leakage. The examination was easily and conclusively effected with the aid of the cylinder relief-valves for the steam-valves and piston, and then, by blocking up the steam-valves, the tightness of the exhaust-valves could be determined by the temperature of the condenser after standing a reasonable time. In no case, after the most thorough and protracted examination, could any appreciable leakage be discovered. The engine would stand for a day with the steam pressure on the valves without raising the temperature of the condenser in the least degree, or discharging any steam through the relief-valves when opened. On taking off the cylinder head the entire cylinder surface was found to have the polish of a mirror. Great care was taken in packing the stuffing-boxes of the valve-stems to keep them in perfect truth; and it is confidently believed that the condition of the engine as regards tightness and the accuracy with which its organs functioned was unexceptionable.

The temperature of the heated gases emerging from among the tubes into the smoke-pipe uptake was ascertained by immersing a high grade mercurial thermometer into a copper pot 3 inches in diameter and 14 inches deep, filled with oil and hung in juxtaposition to the front tubes of the centre furnace of the port boiler. The pot could be easily taken out and the temperature noted without the thermometer falling sensibly. The mean temperature of the gases, thus determined, was 520° Fahr.

The watches, with the exception of a few kept by Chief Engineer Isherwood, were kept by Chief Engineers Zeller, Long, and Stimers, and First Assistant Engineer Wm. H. Rutherford, who personally took all the indicator diagrams, recorded all the observations, and superintended the entire operations of the engine room. The men were in three watches, and so stationed that each had his special duty to do—firing, weighing, pumping, oiling, or tending tank—and only that; and being experienced and reliable they performed their offices in a manner that left nothing to be desired.

From the foregoing, it will be perceived that every precaution was taken in conducting the experiments, and in securing normal conditions for the machinery to obtain exact results. Great care, too, was taken

# TABLE No.

ENGINE OF

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| NUMBER OF LINE. | | | |
|---|---|---|---|
| 1 | TIME. | | Date of Commenc |
| 2 | | | Date of Ending E |
| 3 | | | Duration of the E |
| 4 | TOTAL QUANTITIES. | | Number of Double |
| 5 | | | " Pound |
| 6 | | | " " |
| 7 | | | " " |
| | TEMPERATURE IN DEGREES F. | | " Feed-wate |
| | | | " Engine R |
| | | | " External |
| 41 | TOTAL EVAPORATION. | FROM FEED-WATER | Pounds of Steam the pressu |
| 42 | | | Pounds of Steam duce the t lent, . |
| 43 | | | Pounds of Steam |
| 44 | | | " |
| 45 | | | " |
| 46 | | | " |
| 47 | ECONOMIC EVAPORATION. | | Pounds of Steam by INDIC |
| 48 | | | Pounds of Steam by TANK |
| 49 | | | Pounds of Stea by INDIC |
| 50 | | | Pounds of Stea by TANK |
| 51 | | | Pounds of Stea bustible, |
| 52 | | | Pounds of Stea bustible, |
| 53 | | | Pounds of Ste bustible, |
| 54 | | | Pounds of Ste bustible |
| 55 | CONDEN-SATION. | | Per centum of total po |
| 56 | | | Per centum of INDICA |

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to maintain equality of conditions throughout the whole set of experiments in every respect except the measure of expansion used for the steam, the economic effect due to which was the sole issue to be determined.

The data and results of the experiments will be found in the two following tables, which embrace only those in which all the conditions from beginning to end were such as could satisfy the most hypercritical. These, too, are the experiments which give the maximum effect for the greater measures of expansion employed, and as they are both exact and sufficiently numerous for the solution of the problem, it is useless to add any others which uncontrollable variations in the conditions during their progress could lay open to a doubt.

Before proceeding to a discussion of the results of these experiments, it is proper to give the following explanation of the tables containing them, in order to show the manner in which they are obtained from the data:

### EXPLANATION OF TABLES 1 AND 2 CONTAINING THE DATA AND RESULTS OF THE EXPERIMENTS MADE WITH THE STARBOARD ENGINE OF THE UNITED STATES STEAMER "MICHIGAN" TO DETERMINE THE RELATIVE ECONOMY IN RAPPORT OF FUEL TO POWER OF USING STEAM WITH DIFFERENT MEASURES OF EXPANSION.

In the following two tables will be found the observed data and the calculated results of the experiments made with the machinery and in the manner precedingly described. Table No. 1 contains the exact experimental determinations under the conditions noted. Table No. 2 contains the results of the experiments detailed in Table No. 1, but calculated only for the weight of steam used in rapport to power developed, and corrected for equality of back pressure against the piston—which equality did not obtain in the experiments, but which it is necessary to adopt, in order to show the *true* relative economy of the different measures of expansion employed; for whatever back pressure can be obtained in one case can in any other. And first, of

## TABLE No. 1.

The experiments were seven in number, and the results are arranged in parallel columns under their appropriate headings. In these experiments the steam was cut off at $\frac{11}{12}$ths, $\frac{7}{10}$ths, $\frac{4}{9}$ths, $\frac{3}{10}$ths, $\frac{1}{4}$th, $\frac{1}{8}$th, and $\frac{4}{45}$ths of the stroke of the piston from the commencement. As hereinbefore explained, the nature of the valve-gear did not permit the steam to follow farther than $\frac{11}{12}$ths of the stroke, nor to be cut off at any intermediate point between $\frac{7}{10}$ths and $\frac{4}{9}$ths the stroke, and an inspection of the results will show at a glance the uselessness of an experiment cutting off shorter than the $\frac{4}{45}$ths of the stroke.

For facility of reference the quantities are arranged in groups and the lines containing them numbered.

TIME. Lines 1 and 2 contain the dates of commencing and ending each experiment. Line 3 contains the duration of each experiment in consecutive hours.

TOTAL QUANTITIES. Line 4 contains the total number of double strokes made by the piston, obtained by subtracting the number of the counter at the beginning from the number at the end of each experiment. Line 5 contains the total number of pounds of feed-water pumped into the boilers as determined by the number of tanksfull used, the capacity (70 cubic feet) to which the tank was exactly filled each time, and the weight of a cubic foot of water at the temperature of the feed-water as given on line 38. In making this calculation the weight of a cubic foot of water at 62° Fahr. was taken at 62·321 pounds, and the corrections for variations from this temperature were made according to the results obtained by the elaborate researches of KOPP on the expansion of water.

Line 6 contains the actual weight of coal consumed during the experiment; and line 7 contains the weight of refuse in clinker, ashes, soot, and dust, belonging to this weight. Line 8 contains the remainder after deducting the quantities on line 7 from those on line 6. Line 9 contains the per centum which the quantities on line 7 are of those on line 6.

QUANTITIES PER HOUR. The quantities on lines 10, 11, and 12, are respectively the quotients of the division of the quantities on lines 5, 6, and 7, by the quantity on line 3. Lines 13 and 14 are respectively the quotients of the division of the quantities on lines 11 and 12 by 90, the number of square feet of grate surface in the boilers.

ENGINE. Line 15 contains the quotients of the division of the quantities on line 4 by 4320, the duration of each experiment in minutes. Line 16 contains the mean vacuum in the condenser; and line 17 contains the mean height of the barometer during each experiment.

STEAM PRESSURES. Line 18 contains the mean boiler pressure in pounds per square inch above the atmosphere. Lines 19, 20, 21, and 22 contain the steam pressure in the cylinder in pounds per square inch above the absolute zero, or point of no pressure, at the commencement of the stroke of the piston, at the point of cutting off the steam, at the end of the stroke of the piston, and against the piston during its stroke. These quantities were obtained from the indicator diagrams taken and analyzed as hereinbefore described, and are corrected for the atmospheric pressure as given by the barometer on line 17. Line 23 contains the mean gross effective pressure by indicator in pounds per square inch on the piston during its stroke. This pressure is the mean of the ordinates erected at right angles to the atmospheric line and bounded by the periphery of the diagram. The sum of the quantities on lines 22 and 23 will, of course, be the total average pressure and represent the complete statical effect of the steam upon the piston, part of which is balanced by the back pressure on line 22, part by the pressure of 2·1 pounds per square inch of piston required to work the engine *per se*, leaving the remainder for the net pressure producing the rotation of the wheels.

POWER. Lines 24, 25, and 26 contain the number of horses power of 33,000 pounds raised one foot high per minute, developed by the engine and calculated for different conditions of pressure, but, in the same experiment, for the same speed of piston per minute, namely, the product of the quantity on line 15 and the length, 16 feet, of a double stroke. The gross effective horses power on line 24 is calculated for the mean gross effective pressure on line 23. The total horses power on line 25 is calculated for the sum of the pressures on lines 22 and 23. The net horses power on line 26 is calculated for the pressure on line 23, minus the 2·1 pounds per square inch required to work the engine *per se*.

Lines 27, 28, and 29 contain, respectively, the number of pounds of feed-water consumed per hour per gross effective, total, and net indicated horse power. These weights are the quotients of the division of the quantities on line 10 by the quantities on lines 24, 25, and 26. Lines 30, 31, and 32 contain, respectively, the number of pounds of coal consumed per hour per gross effective, total, and net indicated horse power. These weights are the quotients of the division of the quantities on line 11 by the quantities on lines 24, 25, and 26.

Lines 33, 34, and 35 contain, respectively, the number of pounds of combustible consumed per hour per gross effective, total, and net indicated horse power. These weights are the quotients of the division of the quantities on line 12 by the quantities on lines 24, 25, and 26.

TEMPERATURES. Line 36 contains the temperature of the injection-water. It will be observed that this

temperature was the same in all the experiments, and the uniformity was due to the hot water discharged by the air-pump, which, surrounding the vessel, mingled with the ice-cold water of the calm confined basin in which the vessel floated. It was wholly owing to the hot water thus discharged that the experiments were practicable at this season, as it kept the ice melted for a number of acres around; and it was curious to observe the advance and recession of the sharp line of the ice as the temperature of the atmosphere rose and fell, or as the quantity of water hourly discharged was greater or less. Line 37 contains the temperature of the hot-well. It was the same in all the experiments, care having been taken to maintain it uniform. Line 38 contains the mean temperature of the feed-water in the tank. This water was derived mainly from the hot-well and partly from overboard, as previously described. The proportion of outside water to that from the hot-well was variable, and, in some experiments, a much longer time being required for consuming a tankful, the tank water was more affected by the greater exposure to the engine room temperature. From these causes the temperature of the feed-water in the different experiments varied, though the temperature of the hot-well and of the outboard water continued constant throughout.

Line 39 contains the temperature of the engine room, taken by the thermometer attached to the barometer situated in the centre of the engine room. Line 40 contains the temperature of the external atmosphere, taken by a thermometer suspended outboard. This temperature is 6° Fahr. higher than that of the atmosphere at a distance from the vessel, owing to the vapor rising from the hot water discharged by the air-pump.

Total Evaporation. Line 41 contains the total number of pounds of steam discharged from the cylinder at the end of the stroke of the piston. It is calculated from the weight per cubic foot of the steam of the pressure at the end of the stroke (line 21), as determined by Fairbairn's formula, from the total number of double strokes of piston made (line 4), and from the space displacement of the piston per stroke (56·544 cubic feet), exclusive of the rod, plus the space comprised (3·28 cubic feet) in the steam passage and clearance at one end of the cylinder, due allowance having been made for the back pressure (line 22) already occupying this space when the exhaust-valve closed, which was at 0·4666 foot before the piston reached the end of its stroke. The weight of steam thus calculated was evaporated from the temperature of the feed-water, line 38.

The formula of Fairbairn is as follows, namely: Let v be the specific volume of the steam or volume compared with that of an equal weight of water, and P the pressure in inches of mercury corresponding to the pressure in pounds per square inch of the steam at the end of the stroke of the piston (line 21); then $v = 25{\cdot}62 + \frac{49513}{P + 0{\cdot}72}$; and taking the weight of a cubic foot of water at 62° Fahr. to be 62·321 pounds, the weight of the cubic foot of steam of the pressure P will be $\frac{62{\cdot}321}{v}$.

Line 42 contains the total number of pounds of steam evaporated in the boiler from the temperature of the feed-water (line 38), and condensed in the cylinder to furnish the heat transmuted into the power developed by the engine, according to Joule's equivalent of one pound of water raised one degree of temperature on Fahrenheit's scale for every 772 foot-pounds developed by the engine, which would make the thermal equivalent of one indicated horse power $\left(\frac{33,000}{772} =\right)$ 42·7461 pounds of water raised one degree Fahr. To make the calculation: let $k$ = the number of *total* indicated horses power (line 25) developed by the engine: $e$ = the total heat of steam of the pressure at the end of the stroke of the piston (line 21) in degrees Fahr. according to Regnault: $g$ = the temperature in degrees Fahr. of the same steam: and $t$ = the time in

minutes ($72 \times 60 = 4320$, line 3) during which the indicated horses power $k$ acted: then $\frac{k \times 42{\cdot}7461 \times t}{e - g} =$ the number of pounds of steam on line 42.

Line 43 contains the total number of pounds of steam that would have been evaporated according to indicator measurement, had the temperature of the feed-water been 100° Fahr. instead of that given on line 38. It is the sum of the quantities on lines 41 and 42 corrected for the difference of temperature of feed-water 100° Fahr., and of the temperature on line 38.

Line 44 contains the total number of pounds of steam that would have been evaporated according to tank measurement, had the temperature of the feed-water been 100° Fahr., instead of that given on line 38. It is the quantity on line 5 corrected for the difference of temperature of feed-water 100° Fahr., and of the temperature on line 38.

Line 45 contains the total number of pounds of steam that would have been evaporated according to indicator measurement, had the temperature of the feed-water been 212° Fahr. instead of that on line 38. Line 46 contains the total number of pounds of steam that would have been evaporated according to tank measurement, had the temperature of the feed-water been 212° Fahr. instead of that on line 38. These quantities are obtained in the same manner as those on lines 43 and 44, allowance being made for the increase of temperature from 100° to 212°. In making these calculations the total heat of steam of the boiler pressure has been taken from REGNAULT'S tables.

ECONOMIC EVAPORATION. Lines 47 to 54, both inclusive, contain the number of pounds of steam evaporated per pound of coal, and per pound of combustible, from the temperatures 100° and 212° Fahr. of feed-water, according to both tank and indicator measurement. The evaporation is given from both temperatures for convenience of comparison with the results of experiments on other boilers.

Lines 47 and 51 contain the number of pounds of steam evaporated per pound of coal, and per pound of combustible, from the temperature 100° Fahr. of feed-water by indicator measurement. They are the quotients of the division of the quantity on line 43 by the quantities on lines 11 and 12, respectively.

Lines 48 and 52 contain the number of pounds of steam evaporated per pound of coal, and per pound of combustible, from the temperature 100° Fahr. of feed-water by tank measurement. They are the quotients of the division of the quantity on line 44 by the quantities on lines 11 and 12, respectively.

Lines 49 and 53 contain the number of pounds of steam evaporated per pound of coal, and per pound of combustible, from the temperature 212° Fahr. of feed-water by indicator measurement. They are the quotients of the division of the quantity on line 45 by the quantities on lines 11 and 12, respectively.

Lines 50 and 54 contain the number of pounds of steam evaporated per pound of coal, and per pound of combustible, from the temperature 212° Fahr. of feed-water by tank measurement. They are the quotients of the division of the quantity on line 46 by the quantities on lines 11 and 12, respectively.

CONDENSATION. Line 55 contains the per centum which the quantity on line 42 is of the quantity on line 5. It expresses that per centum of the heat imparted to the steam in the boiler, which is theoretically utilized in producing the total dynamic effect developed by the piston of the engine.

Line 56 contains the per centum which the difference between the sum of the quantities on lines 41 and 42, and the quantity on line 5, is of the latter quantity. It expresses, in per centum of the total quantity of feed-water pumped into the boilers, the difference between the weights of steam used, as determined by indicator and tank measurement, including in the indicator measurement the weight of steam condensed to furnish the heat transmuted into the total power developed by the engine.

The causes of this discrepancy may be numerous. If the boilers lose water by leakage, by priming, or

# TABLE No. 2:—(

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| NUMBER OF LINE. | | | |
|---|---|---|---|
| 1 | STEAM PRESSURE. | IN CYLINDER PER INDICATOR. | In pounds per square inch al |
| 2 | | | " " " |
| 3 | | | " " " |
| 4 | | | " " " |
| 5 | | | Mean gross effective pressur stroke, . |
| 6 | | | Pressure in pounds per squa: *per se*, . |
| 7 | | | Mean net effective pressure : |
| 8 | | | Per centum which the mean sure, . |
| 9 | POWER. | ABSOLUTE. | Gross Effective Horses Pow |
| 10 | | | Total Horses Power develop |
| 11 | | | Net Horses Power usefully |
| 12 | | ECONOMIC. | Pounds of Steam consumed |
| 13 | | | " " |
| 14 | | | " " |
| 15 | COMPARATIVE. | | Speed of the Piston, in feet |
| 16 | | | Comparative bulks of Stean |
| 17 | | | Mean total pressure on Pist |
| 18 | | | " " |
| 19 | | | Comparative economic resu |
| 20 | | | " " Power, . |
| 21 | | | Comparative economic resu |
| 22 | | | " " |
| 23 | | | Comparative capacity of Power, . |
| 24 | | | Difference, due to all cause the Boilers, accordin from the Cylinder i Piston, per Indicato |

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by passing it over to the cylinder in the vesicular state, the quantity thus lost will be included. If the cylinder-valves or piston leak steam to the condenser, the quantity thus leaked will be included. If steam be condensed in the steam-pipe, valve-chests and cylinder, from any causes whatever other than the production of the power, and if a portion of the water formed by this condensation be re-evaporated in the cylinder, then the difference of the weights condensed and re-evaporated will be included.

If, however, there be no water leaked or passed over from the boilers; and no steam leaked by the valves or piston of the cylinder; and neglecting any condensation in the steam-pipe and valve-chests due to external refrigeration; and supposing no re-evaporation in the cylinder; then the discrepancy would be due wholly to condensation in the cylinder from all causes other than the production of the power. But admitting such condensation, the principal part of which, when the steam is used expansively, must take place in the first portions of the stroke of the piston under the higher pressures and temperatures there existing, some re-evaporation would inevitably take place in the last portions of the stroke when the pressures and temperatures had been greatly reduced by the expansion; and this re-evaporation is measured by the indicator, for it forms part of the pressure in the cylinder at the end of the stroke of the piston. The quantity then, on line 56, would by no means express the whole condensation in the cylinder due to other causes than the production of the power—even supposing no loss of water from the boilers, or of steam by leakage past the cylinder-valves and piston to the condenser, or by external refrigeration of steam-pipe and valve-chests—but only the excess of that condensation over the re-evaporation. The real condensation in the steam-pipe, valve-chests, and cylinder, always exceeds, and may greatly exceed that expressed on line 56. The heat for re-evaporation is, of course, in the metal of the cylinder, and in the water of condensation itself; the temperature of the metal, and of the deposited water, during the first portions of the stroke, being higher than the boiling point of the water under the lessened pressure caused by the expansion during the last portions. The cylinder, in fact, performs the double functions of a condenser and a boiler upon the steam side of the piston during the stroke.

## TABLE No. 2.

The quantities given in table No. 1 are the precise ones obtained by experiment; but some of them require to be slightly corrected, for the purpose of properly making exact comparisons. The quantities to be corrected are only those of the mean gross effective pressure on the piston, and back pressure against it (Table No. 1, lines 22 and 23), together with those of the mean gross effective, total, and net horses power depending on them (Table No. 1, lines 24, 25, and 26). These corrections are made necessary by the fact of the inequality of the back pressure during the experiments; but, as it was caused by such accidental circumstances as air leakages, different proportion of cylinder steam-port to weight of steam discharged at the end of the stroke of the piston, and different speed of piston, all of which can be made the same and the back pressure rendered equal, it is necessary for a proper comparison to make the back pressure the same in all the experiments, and to rectify the mean gross effective and net pressures on the piston in accordance with this equality. This has been done in Table No. 2, which contains, in addition, all the other data and calculated results requisite to a complete determination of the relative economy due to the different measures of expansion. In this table, as in Table No. 1, the quantities have been grouped and the lines numbered for convenience of reference.

Steam Pressure. Lines 1, 2, and 3 contain the same quantities as lines 19, 20, and 21 of Table No. 1. They are respectively the pressures in the cylinder by indicator, in pounds per square inch above zero,

at the beginning of the stroke, at the point of cutting off the steam, and at the end of the stroke of the piston. Line 4 contains the corrected back pressure above zero, in pounds per square inch, against the piston during its stroke. The quantity 2·7 pounds has been adopted for this pressure because it is the least given during the experiments; and as with equal initial cylinder pressures the results are more unfavorably affected by back pressure as the steam is used more expansively, it was proper to accept the least practicable; and as whatever back pressure could be obtained in one case could be obtained in all, this minimum must be applied throughout. Further, a back pressure of 2·7 pounds per square inch of piston may be regarded as the minimum with any engine. The average with steam-engines under the conditions of ordinary practice is about 4 pounds, which, if adopted, would make the economic results much more unfavorable as the steam was used with the greater measures of expansion.

Line 5 contains the mean gross effective pressure per square inch that would have been on the piston had the back pressure against it been the quantity on line 4. It is the difference between the mean total pressure on the piston in pounds per square inch, line 18, and the back pressure of 2·7 pounds.

Line 6 contains the pressure required per square inch of piston, by indicator, to work the engine *per se.*

Line 7 contains the mean net effective pressure on the piston, in pounds per square inch, that would have obtained had the back pressure against it been the quantity on line 4. It is the difference between the mean gross effective pressure that would have been on the piston had the back pressure against it been the quantity on line 4, and the pressure of 2·1 pounds per square inch of piston required to work the engine *per se.*

Line 8 is the per centum which the quantity on line 7 is of the quantity on line 18. It shows the proportion of the mean total pressure on the piston that would be utilized in rotating the wheels, were the mean effective pressure the quantity on line 7.

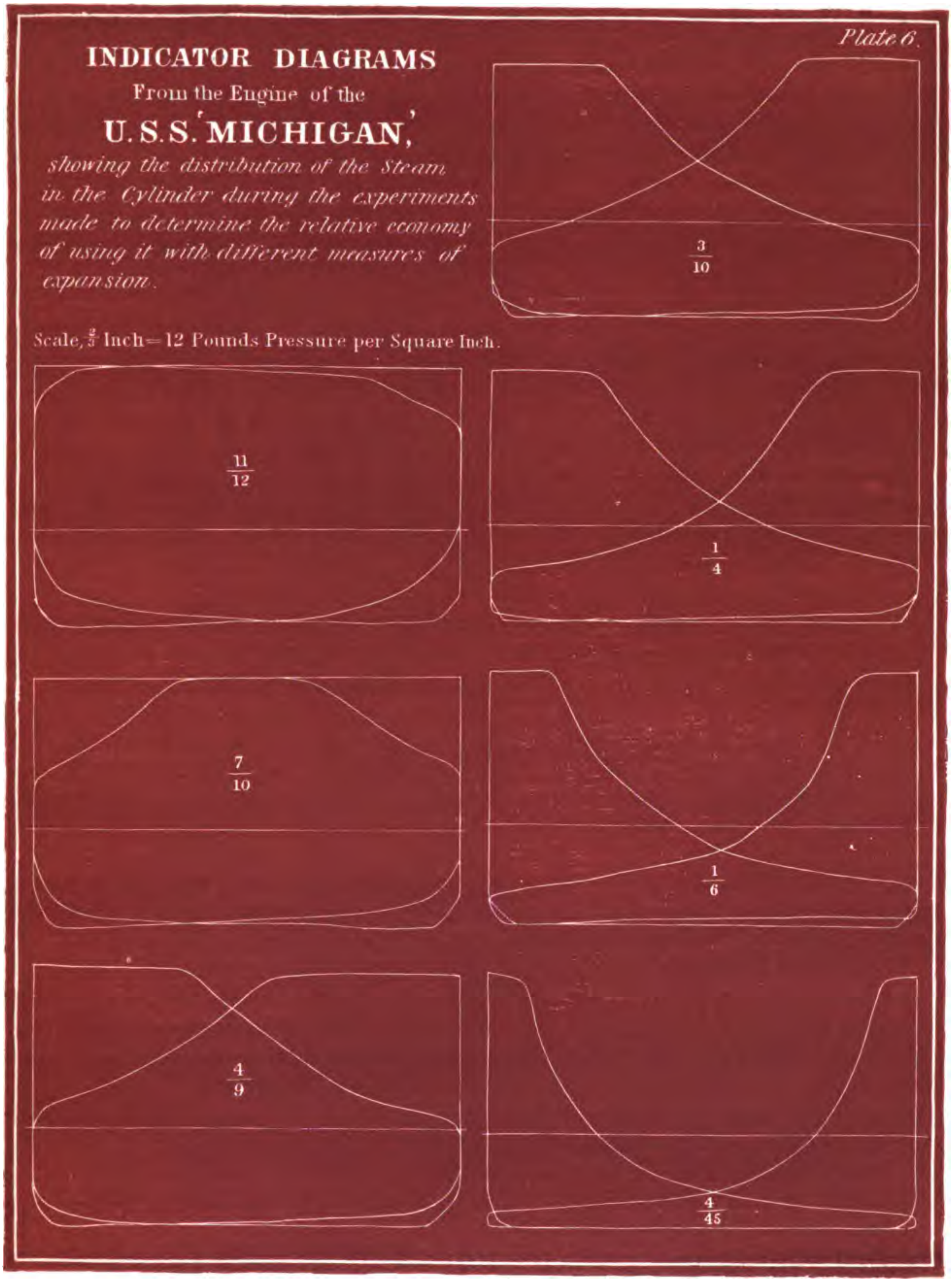
POWER, ABSOLUTE. Lines 9, 10, and 11 contain, respectively, the gross effective, total, and net indicated horses power developed by the engine. They are calculated for the piston pressures on lines 18, 5, and 7, and for the speed of piston on line 15.

POWER, ECONOMIC. Lines 12, 13, and 14 contain, respectively, the number of pounds of feed-water consumed per hour to produce the gross effective, total, and net indicated horses power developed by the engine. They are the quotients of the division of the quantity on line 10 of Table No. 1, by the quantities on lines 9, 10, and 11.

COMPARATIVE. Line 15 contains the speed of the piston in feet per minute. It is the product of the quantity on line 15 of Table No. 1, and the length, 16 feet, of a double stroke of the piston.

Line 16 exhibits, comparatively, the bulks of steam withdrawn from the boilers in equal times during the experiments. It serves—as the steam room of the boilers was constant—to convey an idea of the more unfavorable condition that existed, when the steam was used with less expansion, for obtaining it dry from the boilers: and of the probability of greater loss in these cases from priming, or the passage over of boiler water in the vesicular state, supposing either to have occurred. It is proper to state, however, that in none of the experiments could the least evidence of priming or of working over water be discovered, though carefully searched for.

Line 17 contains the mean total pressure, or pressure above zero, on the piston, in pounds per square inch, that should exist according to the law of MARIOTTE. It is calculated for the experimental conditions of the steam comprised in the space between the piston at the end of its stroke and the exhaust-valve, and of the cylinder pressures at the beginning of the stroke and at the point of cutting off the




**INDICATOR DIAGRAMS**
From the Engine of the
**U. S. S. 'MICHIGAN,'**
*showing the distribution of the Steam in the Cylinder during the experiments made to determine the relative economy of using it with different measures of expansion.*

Scale, $\frac{2}{3}$ Inch = 12 Pounds Pressure per Square Inch.

steam (lines 1 and 2). By comparing this quantity with the quantity on line 18, which is the mean total pressure on the piston in pounds per square inch by indicator, a remarkable coincidence will be found; in fact, they agree within the limits of error due to the most accurate practical determination. Nevertheless, the agreement can be only a coincidence, for, in order that the one should be a consequence of the other, it would be necessary that neither condensation from any cause, nor re-evaporation, should have occurred in the cylinder from the beginning to the end of the stroke of the piston, and that the steam should have expanded with pressures precisely in the inverse ratio of the spaces occupied.

Line 19 exhibits, comparatively, the economic result that should have been obtained, with the different measures of expansion, used for the steam, according to MARIOTTE'S law. The calculation is made for the total horses power that would have been developed with the experimental speed of piston on line 15 and the mean total MARIOTTE pressure on line 17, and for weights of steam expended corresponding to the bulks of cylinder filled per stroke of piston up to the experimental point of cutting off, including the space comprised between the piston at the end of its stroke and the exhaust valve, with steam of the experimental pressure on line 2.

Lines 20, 21, and 22 contain the comparative economic results experimentally obtained with the different measures of expansion used. They are calculated, respectively, for the gross effective, total, and net indicated horses power on lines 9, 10, and 11, and for the experimental consumption of feed-water per hour for these powers as given on lines 12, 13, and 14. In fact, the quantities on lines 20, 21, and 22, are respectively the quantities on lines 12, 13, and 14, expressed proportionally, taking those in the first column for unity.

In order to appreciate the difference in the economic results as predicted by the MARIOTTE law, and as given by experiment, the quantities on lines 19 and 21 must be contrasted. These quantities are strictly comparable, and unaffected by either back or friction pressures. They show, in both cases, the entire dynamic effect in ratio to its cost.

Line 23 exhibits, comparatively, the capacity of cylinder that would be required to develop equal net horses power in equal times with equal speed of piston and the net effective pressures on line 7. If we suppose equal length of stroke in the different cases for the piston, its area will be comparatively as these quantities.

Line 24 shows the difference between the weight of feed-water pumped from the tank into the boilers, and the weight of steam discharged from the cylinder into the condenser at the end of the stroke of the piston, as ascertained from the pressure at the end of the stroke per indicator. This difference is that which is due to all causes including the condensation to furnish the heat transmuted into the total power developed by the engine. Now, as there was no leakage of water from the tank, or boiler, or their connexions, or of steam past the cylinder-valves and piston, the whole of this difference, except the insignificant quantity condensed by external refrigeration in the steam-pipe and valve-chests, must be the excess of the weight of steam condensed in the cylinder over the weight re-evaporated from its interior surfaces. The quantity on line 24 is the sum of the quantities on lines 55 and 56 of Table No. 1. By taking it into consideration when comparing the economic results (for total powers) that should have been obtained according to the law of MARIOTTE, line 19, with those obtained by experiment, line 21, a very clear idea will be had of the great antagonistic cause that neutralizes and reverses the economy promised by the purely abstract conditions on which that law is founded.

## DISCUSSION OF THE RESULTS.

In examining the preceding two tables, it will be observed that particular care was taken to have the initial cylinder pressure (Table 2, line 1) the same in all the experiments as nearly as practicable, which, with proper area of conduit open in proportion to quantity of steam used in equal time, would necessarily make the boiler pressure equal. In fact, throughout the experiments the boiler pressure (Table 1, line 18) was nearly equal, the difference being too slight to be of any practical importance. That this is a proper condition for the purpose of the experiments, will be obvious when it is considered that amount of pressure is purely a question of boiler, and not at all one of engine.

If a given power be required to be developed by the same piston, working at a given speed, but with different measures of expansion for the steam, it is plain that the initial pressure must be increased as the steam is used more expansively, and the condition of equality of initial pressure could not obtain. This, however, would only be unnecessarily employing too large a cylinder when the steam was used less expansively; or, in other words, it would be an engineering blunder. The proper method is—for equality of power and of initial pressure—to proportion the cylinder to the measure of expansion adopted for the steam, making its capacity, for equal speed of piston, inversely (Table 2, line 22) as the net effective pressures upon the piston. (Table 2, line 7.) From the justness of these premises there is no escape, and it is preposterous to base a claim of economy for large measures of expansion, not upon the expansion *per se*, but upon higher initial pressure, when that same pressure can be employed just as easily and well when using steam without expansion, if the cylinder be properly proportioned to the work. Therefore, in making a set of experiments to determine the *practical* economic results of using steam with different measures of expansion, it is an *essential* condition that the initial cylinder pressure be maintained the same in all cases.

It is now proper to give the reasons why a high initial pressure is desirable in view of the economical production of the power. And, first, of the generation of the steam.

As the dynamic effect of a given weight of steam increases in a higher ratio than the heat required to evaporate it—owing to the accompanying increase in the temperature of the steam—it is obviously desirable to use it in the cylinder at the maximum pressure throughout the entire stroke. We say *use* it in the cylinder, for the economy in function of pressure *per se*, attaches to the pressure under which the steam is used, and not to that under which it is generated, because its dynamic effect is developed during its use in the cylinder, and not during its generation in the boiler. Now, it is plain that, starting with the same initial cylinder pressure, the steam will be used with the highest pressure throughout the stroke of the piston, when it is used without expansion; and the more it is expanded, the more is the pressure reduced in the cylinder, and the advantage lost that attaches to higher pressure *per se*. This is one *practical* point of gain for smaller measures of expansion over larger ones—a point entirely ignored when the problem is merely theoretically considered, but which must be included in a practical determination, because it is an inseparable function of the physical laws of steam.

But there are other reasons, far more potent than the above, for desiring a high pressure on the piston throughout its stroke, whether the steam be used with or without expansion. They are to be found in the fact of the imperfect manner in which the boiler pressure is applied in the practical steam engine. Could the condensation of the steam, after the stroke of the piston is performed, be accomplished under *theoretical*

conditions, that is to say, instantaneously and totally on the opening of the exhaust-valve, leaving no back pressure of vapor in the cylinder to resist the succeeding stroke of the piston, and did the organs of the engine function without friction or other useless resistance, then, neglecting the effect of higher pressure *per se*, it would be a matter of indifference as regards economy of fuel, whether the steam be used at one pressure or at another. These conditions, however, are *practically* impossible; and as the condensation, instead of being instantaneous and total, is slow and partial; and as the organs of the engine have a friction, &c., which require a considerable pressure to overcome; and as, *ceteris paribus*, those back and friction pressures (Table 2, lines 4 and 6) are constant, be the mean total pressure (Table 2, line 18) in the cylinder what it may; and as the same *pro rata* expenditure of steam is required to overcome those useless resistances as to overcome the resistance of the load, it follows, first, that of the total pressure on the piston only that which remains after deducting the back and friction pressures is usefully applied; and second, that as these pressures are constant, there will be a higher proportion (Table 2, line 8) of the total pressure utilized when the latter is higher than when it is lower; and as, with equal initial cylinder pressure, the average total pressure must be less and less the more expansively the steam is used, it is obvious that the greater the expansion, the more will be the difference between the theoretical and practical results (Table 2, lines 19 and 22) due to these causes. From these considerations it is plain that, starting with even the maximum practicable initial pressure in the cylinder, the expansion may be carried so far as to make so large a portion of the total average pressure to be consumed in merely overcoming the back and friction pressures, that the economy of the same weight of steam used with the same initial cylinder pressure, but without expansion, shall greatly exceed the economy of the expanded steam. This result would be impossible under abstract theoretical conditions, which make the economy of expanded steam greater and greater as larger measures are used, up to the point where the attraction of the particles of steam for each other would balance the repulsive force of the heat. This illustration, alone, shows how wide a difference there must be between the theoretical and practical results, due purely to the differential of the total pressure and the sum of the back and friction pressures, and conclusively proves how futile all determinations of gain must be for the use of steam expansively, made by processes which ignore antagonistic conditions of such magnitude.

In examining the speed of piston (Table 2, line 13) during the different experiments, it will be observed that it was neither equal nor relatively what was due to the mean net effective pressure (Table 2, line 7) upon it, supposing the same paddle surface with the same immersion to be employed, and the resistance encountered from the water to be in the ratio of the square of the velocity of that surface. The speed of piston, however, is a condition that (neglecting the refrigerating influence of the condenser temperature hereafter considered) does not affect the economic comparison, which, then, is governed solely by the pressures. For it is evident that, with any given degree of expansion, a given weight of steam is used per stroke, and a given mean pressure obtained without any regard whatever to the time in which the stroke is performed; and as the former expresses the cost and the latter the result, the economy is not affected by the speed of piston, or number of strokes made per minute.

With regard to the influence exerted by the speed of the piston on the condensation of steam due to the exposure of the interior surfaces of the cylinder alternately to the temperature on the steam and exhaust sides of the piston, it will be perceived that, as these surfaces are exposed half the time to the steam temperature and half the time to the exhaust temperature, and as, *ceteris paribus*, this condensation is in direct ratio to the time, it will be the same absolute quantity for equal time, be the number of strokes made in that time, or, in other words, the speed of the piston, what it may. But as the weight of steam used in

equal time through the cylinder, is in the direct ratio of the number of strokes of piston made during that time, it is manifest that, *proportionally to this weight*, the quantity of steam condensed by this refrigerating influence will be inversely as the number of double strokes made by the piston in a given time, that is, inversely as its speed. For instance: doubling the speed of piston will halve—not the absolute weight of steam condensed due to the cause in question, for that will remain constant—but the per centum which that absolute weight is of the total weight of steam passed through the cylinder in equal time. Thus, if the condensation were, with 20 strokes of the piston per minute, 1 per centum of the total weight of steam passed through the cylinder during that minute, it will be, with 10 strokes of the piston per minute, 2 per centum of the total weight passed through the cylinder per minute to make those 10 strokes.

The condensation in the cylinder due to the variable *temperatures* of its metal, caused by the alternate exposure of its interior surfaces to the different temperature of the steam on the opposite sides of the piston, is too insignificant to be included in a practical estimate, even under the most favorable conditions. The surfaces in question are, of course, the sides, ends and nozzles of the cylinder, the interior of the valves, and the disc of the piston. To understand how very small the condensation due to this cause must be, we will consider the conditions of the simplest case, namely, that which occurs when using the steam without expansion. For this purpose, let us suppose the piston to have just arrived at one end of its stroke, and the whole interior of the cylinder to be filled with steam, of the boiler temperature, and its surfaces, to a certain depth, to have the same temperature. Now, let the exhaust-valve be opened, and then this steam will be discharged into the condenser and replaced with vapor of the greatly less temperature of the back pressure. This vapor will, of course, absorb heat from the metal of the cylinder, but the maximum quantity can only be that which would raise the temperature of the cylinder full of back pressure vapor to nearly that of the metal; and if we consider the extreme tenuity of this vapor, the trifling weight of a cylinder full, and the difficulty with which it absorbs heat, we shall appreciate how little will be taken up. In the practical operation of the steam-engine, the cylinder full of back pressure vapor is pushed out by each stroke of the piston into the condenser, and, of course, carries with it whatever heat it had obtained from the metal of the cylinder by contact and by radiation. That the quantity, however, is practically inappreciable, will appear from an examination of the experiment made with the steam cut off at $\frac{11}{12}$ths of the stroke of the piston from the commencement, in which the whole difference between the weight of feed-water pumped from the tank into the boilers, and the weight of steam accounted for by the indicator, is only 2·91 per centum of the former. In this slight discrepancy is, of course, included the loss from every kind of leakage, and from the condensation by external refrigeration in the steam-pipe, valve-chests and cylinder. Slight as the loss from this particular cause is seen to be when using the steam without expansion, it will be still less when the steam is used expansively, decreasing as the measure of expansion is increased; for as the temperature of the steam urging the piston, will continue to fall from the point of cutting off to the end of the stroke, whatever heat the steam of reducing temperature obtains from the metal of the cylinder, previously imparted by its higher temperature before the point of cutting-off, will be utilized in producing a dynamic effect upon the piston, and the temperature of the metal will, to that degree, be made lower for the back pressure vapor to act on, and which will therefore obtain less heat from it.

Apart from the above, there may, however, under favorable conditions, be a very considerable transference of heat from the metal of the cylinder to the condenser; but it is due to a cause entirely independent of the *temperature* of the back pressure vapor. This cause is the absorption of heat from the metal by the re-evaporation of the water deposited on it by previous condensation of steam.

Let us suppose the temperature of the metal of the cylinder to be less than that of the entering steam during the first parts of the stroke of the piston, a condensation of this steam will take place by its impartation of heat to equalize these temperatures. The steam, the metal of the cylinder, and the water of condensation deposited on that metal, will then have the same temperature. If, now, during the last part of the stroke, the steam pressure be lessened from any cause, as throttling, or by expansion, the water of condensation and the metal of the cylinder will have a higher temperature than the boiling point of water beneath the reduced pressure. Under these circumstances the water will re-evaporate, and the evaporation will continue as long as there is water and a greater temperature than its boiling point either in itself or in the metal on which it rests; and as this evaporation renders latent an enormous quantity of heat, the temperature of the metal—supposing a sufficiency of water—can be reduced to that of the steam at the end of the stroke of the piston; and, if there be a superabundance of water, the evaporation can still continue after the opening of the exhaust-valve, and the temperature of the metal be further reduced to that of the back pressure. By means of this alternate process of condensation and re-evaporation in the cylinder, the temperature of its metal may be raised and lowered the difference of the initial steam and back pressure during each stroke of the piston. This change of temperature in the metal of the cylinder due to the impartation of heat by the steam in the act of condensation, and to the abstraction of this heat by re-evaporation, is confined to a certain depth, depending on the conductibility of the metal and other conditions; beneath this depth the temperature will be uniform, and the mean due to the variations in the cylinder.

The loss of heat, due to the causes just stated, will be greater with greater measures of expansion; because, starting with equal initial pressure, the more expansively the steam is used, the greater will be the reduction of pressure; and this reduction is not only the efficient cause of the evaporation, but it graduates the quantity by determining the minimum boiling point of the water. Further, the greater the measure of expansion used, the greater will be the surface in equal time from which evaporation takes place.

The loss is affected, likewise, by the absolute pressure at the point of cutting off; for, with the same measure of expansion, the difference between the temperatures of the steam at the point of cutting-off and at the end of the stroke of the piston increases with increase of pressure. Also, as the back pressure remains constant let the pressure at the point of cutting-off be what it may, the minimum boiling point will remain constant, too, for the water not evaporated before the end of the stroke of the piston; but the temperature of the steam and of the metal of the cylinder being higher with greater pressures during the first part of the stroke, they will have more heat stored in them to be expended in re-evaporation to the boiling point due to constant back pressure, and to be re-supplied by the condensation of steam during the first part of the succeeding stroke.

It is impossible to ascertain the absolute amount of loss from the causes in question, but it obviously may be considerable, and it certainly will be greater and greater as the steam is used more and more expansively.

It will thus be seen that the transference of heat from the metal of the cylinder to the condenser, due to the variable pressures and temperatures in the cylinder during a stroke of the piston takes place in two separate ways entirely independent of each other, and differently influenced by changes in the measure of expansion used for the steam.

1st. There is the loss of heat due purely to difference of temperature between the metal of the cylinder and the vapor of the back pressure. In this case, the heat being abstracted from the metal by the vapor in the same manner as a gas would absorb it, becomes to the extent of that abstraction superheated;

and as a cylinder full is pushed into the condenser by each stroke of the piston, it carries with it the quantity of heat measured by this superheating, *the whole of which goes to the reduction of the economic effect produced by the fuel.* This cause acts independently of pressure, is distinct from that producing re-evaporation of the water of condensation, and, *ceteris paribus,* is attended with a less loss the more expansively the steam is used.

2d. There is the loss due to the heat absorbed from the metal of the cylinder and rendered latent by the re-evaporation of the water formed by the condensation of steam, whenever the pressure is reduced by throttling or expansion. In this case, the temperature of the metal of the cylinder at the commencement of the stroke of the piston being less than that of the entering steam, a condensation of the latter occurs to furnish a sufficient quantity of heat to equalize them, when the temperature of the steam, of the metal and of the water of condensation deposited on it, will be the same. This state of things continues up to the point at which the steam is cut off: but from the moment expansion commences, and purely as a result of the lessened pressure it produces, a re-evaporation of this water of condensation commences and continues as long as the pressure continues to lessen, not only up to the end of the stroke of the piston, but after the opening of the exhaust-valve if any water then remain. Besides this continuous re-evaporation, there is also a continuous condensation during the expansion part of the stroke of the piston; for as the piston recedes, it continues to expose portions of the cylinder having a less temperature than that of the steam expanding upon them. The heat producing this re-evaporation is derived from the water itself, and from the metal of the cylinder on which it rests, and whatever quantity is thus abstracted from the metal during one stroke of the piston, must be re-supplied by the steam during the next. These effects are independent of the *temperature* of the steam in the cylinder, and are due purely to the continuous reduction of *pressure* by means of which the temperature of the metal and water are always above the boiling point of water normal to the pressure upon them. The economic result of the fuel is greatly affected by this cause. In the first place, all the heat absorbed from the metal of the cylinder, while the exhaust-valve is open, is a total loss of the fuel required to generate it. In the next place, as regards the heat abstracted from the metal of the cylinder during the stroke, only a portion of it causes a loss of fuel; because, the re-evaporated steam produces a useful dynamic effect upon the piston. But, as a certain time must elapse between the condensation and the re-evaporation in the cylinder, and as during that time the piston continues its movement, the whole dynamic effect of the steam due to the time it remains water, is lost. In short; every particle of the steam entering the cylinder must, in order to produce its full dynamic effect, retain the vaporous form from the beginning to the end of its stroke; and if there be any portion of the stroke during which it loses the form of vapor, a dynamic effect measured by that portion and the wanting pressure, is lost.

It will be perceived by examining line 56 of Table No. 1, that the difference between the weight of feed-water pumped into the boilers and of steam accounted for by the indicator, which was but trifling when the steam was cut off at $\frac{11}{12}$ths of the stroke of the piston from the commencement, increased rapidly for the $\frac{7}{10}$th and $\frac{5}{8}$th points of cutting off, then remained nearly the same for the $\frac{3}{10}$ths, $\frac{1}{4}$th and $\frac{1}{8}$th points of cutting off, but increased again for the $\frac{4}{45}$th point. The quantities on this line are the differences between the re-evaporation in the cylinder while the exhaust-valve is closed, and the total condensation due to all causes, other than the production of the power, in the steam-pipe, valve-chests and cylinder. We do not pretend to say that the causes we have enumerated are all which have contributed to produce condensation, but we have experimentally determined its aggregate net value—that is, the value after deducting the re-evaporation—let it have been produced by what causes it may, and we find it not only to increase with the measure employed for the expansion, but to reach from nearly nothing when the steam is used without expansion,

to the enormous quantity of 37·16 per centum of the total weight of feed-water pumped into the boilers when the steam is cut off at $\frac{4}{45}$ths of the stroke of the piston from the commencement.

There remains, lastly, to be noted one other cause in the practical steam-engine operating to reduce the economic effect of the fuel, and unequally for different measures of expansion. It is due to the fact that between the end of the cylinder and the piston at the commencement of its stroke, there intervenes the constant space comprised in the clearance and nozzles, and which requires, when the exhaust-valve closes precisely at the end of the stroke, to be filled with steam of the initial pressure less the back pressure, the whole of which, when the steam is used without expansion, is exhausted into the condenser at the end of the stroke without having produced any dynamic effect upon the piston; its use being only to transmit the initial pressure across this space. But when the steam is used expansively, the quantity comprised in this space produces a dynamic effect which becomes *absolutely* greater and greater as the measure of expansion is increased; but as the *proportion* which this quantity of steam bears to the total quantity used increases with increase of expansion in a higher ratio than the value of its dynamic effect, the cause in question acts to lessen the economic result from the fuel as the measure of expansion is increased.

In the case of the experiments, the following table will show the difference in the loss due to the clearance and nozzle space at the end of the cylinder. Line 1 contains the net effective pressure, in pounds per square inch of piston, according to the law of Mariotte under the experimental conditions. It is obtained by subtracting the sum of the back and friction pressures, lines 4 and 6 of Table No. 2, from the total pressure on line 17 of the same table. Line 2 contains what would have been the net effective pressure according to the same law, had the piston come in mathematical contact with the end of the cylinder, and had the same weight of steam with the same initial pressure been used, per stroke, in which case the measure of expansion would be correspondingly lessened. Line 3 contains the difference between the quantities on lines 1 and 2. And line 4 contains the per centum which the quantity on line 3 is of the quantity on line 2.

| NUMBER OF LINE. | FRACTION OF THE STROKE OF PISTON COMPLETED WHEN THE STEAM WAS CUT OFF. | | | | | | |
|---|---|---|---|---|---|---|---|
| | $\frac{11}{12}$ | $\frac{7}{10}$ | $\frac{4}{9}$ | $\frac{3}{10}$ | $\frac{1}{4}$ | $\frac{1}{6}$ | $\frac{4}{45}$ |
| 1 | 29·7 | 26·5 | 22·8 | 18·3 | 16·4 | 12·4 | 7·8 |
| 2 | 29·9 | 26·9 | 23·9 | 19·8 | 17·8 | 14·1 | 9·1 |
| 3 | 0·2 | 0·4 | 1·1 | 1·5 | 1·4 | 1·7 | 1·3 |
| 4 | 0·07 | 1·49 | 4·60 | 7·58 | 7·86 | 12·06 | 14·29 |

An inspection of line 4 of the above table will show how rapidly the loss due to the clearance and nozzle space, increases with the measure of expansion; and how large a proportion it is of the total fuel when the steam is cut off shorter than about half stroke.

There remains for consideration the measures of power and of its cost proper to be adopted for the pur-

pose of justly ascertaining the relative economy of the different measures of expansion. The problem, as regards power, admits of division, and, consequently, of two determinations. The one takes the total horses power developed by the engine (line 10, Table No. 2); the other takes the net horses power usefully applied (line 11, Table No. 2). The former is the entire dynamic effect produced by the steam, and includes the overcoming of all resistances useful or useless. The latter is that portion of the entire dynamic effect which is usefully applied to rotating the wheels, and is the only power that possesses a commercial or practical value. If we wish, therefore, to compare the absolute power which can be derived from a given weight of steam used in a steam-engine under certain conditions of pressure, temperature, and expansion, with the power that can be derived from it under other conditions of pressure, temperature, and expansion, or with a theoretical result, the total power is the measure to use. But if we wish to ascertain the commercial value of the work we desired to have done, the proper measure is the net horses power developed, for that equilibrates and expresses it. It is obvious that it is the cost of only the useful effect produced which is required for a true practical determination, and not the cost of producing other effects which, though inseparable, are neither of value nor desired. In the problem, therefore, of determining experimentally the true relative economy of using steam with different measures of expansion, the net horses power must be accepted as the measure of the effect produced, taking care that the useless, but inseparable, resistances to be overcome shall be the minimum which the properties of matter and the essential organization of a steam-engine will permit; and that the other conditions shall be as favorable for maximum results as it is possible to command in practice; but especially that all conditions shall be the same, save the measure of expansion whose effect is to be determined.

The gross effective horses power developed by the engine (line 9, Table No. 2) is frequently employed for the measure of the useful effect produced, but improperly so, as it includes the power required to work the engine *per se*, which is certainly no more a part of the useful effect than the back pressure overcome. Neither can it be employed for a measure of power, considered philosophically, for it excludes the back pressure, which is as much a part of such power as any other resistance overcome by the engine. A knowledge of the gross effective horses power, however, is useful; for as its cost is affected by the initial cylinder pressure of the steam; by the measure of expansion with which the steam is used; by the goodness of the proportions and setting of the cylinder valves; by their tightness, and the tightness of the cylinder and piston; and by the tightness of the condenser and air-pump and the goodness of their proportions; it is a measure of the aggregate excellence of all these conditions, and of that only. Accordingly, the gross effective horses power developed by the engine in the different experiments, and the cost of the same, will be found in the appropriate groups containing the total and net horses power and their cost.

With regard to the cost of steam power, there are three different measures in use, all of which will be found employed in Table No. 1, namely: the weight of coal; the weight of combustible; and the weight of water by tank measurement, consumed per hour per horse power. The last alone is employed in Table No. 2. Of these, the first is the least exact; the second is more exact than the first, because it eliminates the ashes, the amount of which is an accidental quantity; but the last eliminates everything connected with the generation of the steam, and is, therefore, critically exact, and should be accepted as the true, universal, and only measure of the cost of the dynamic effect, *per se*, produced by the steam in the cylinder.

The weight of coal, under the most favorable circumstances, can only be considered as an indirect and comparative measure of the weight of steam consumed; but, even comparatively, it is not exact unless

all the conditions of boiler and coal continue precisely the same, which is a manifest impossibility, as the calorific effect obtained from the same coal varies with the skill and care of the fireman; with the quantity of water mechanically present in it; with the temperature of the air entering the ash-pit; with the rapidity of the combustion; with the thickness of fuel on the grates; with the hygrometric and barometric conditions of the atmosphere; with the more or less copious supply of air to the furnaces; with the temperature of steam in the boilers; and with the per centum of refuse obtained from the coal; not to include the uncertainty in the determination of the precise weight of coal which generated the steam used in the engine. The weight of combustible is, indeed, a more exact measure than the coal, but only to the extent of the elimination of the error due to difference of refuse.

The weight of feed-water, which is the weight of steam entering the steam-pipe, is unaffected by the causes of error inseparable from the coal or combustible measure; and it furnishes not only a positive and direct measure of the cost of the power, but also an exact comparative measure for the fuel, as, *ceteris paribus*, equal weights of coal generate equal weights of steam; and if the evaporative efficiency of the boiler be known with any kind of coal, the positive weight of that coal to produce the power can easily be obtained from the weight of feed-water. In adopting the weight of feed-water for the proper measure of the cost of the power it is, of course, understood that there must be no water-leakage from the tank, boiler, and their connexions, and no leakage of steam either from the boiler or past the cylinder-valves and piston; for if there be, it will not only cease to be an exact positive measure, but it will not be even an exact comparative one when different weights of steam are used in the same time, because the quantity lost by leakage would be constant. The error caused by steam-leakage equally affects the feed-water and the fuel measures; but that caused by water-leakage affects the former far more than the latter, owing to the fact that, with the feed-water measure, the error will be directly as the leakage, while with the fuel measure, it will be only to the extent of the sensible temperature imparted to the leakage.

Accepting, then, the net horses power developed by the engine as the measure of the effect produced, and the weight of feed-water consumed per hour as the measure of the cost in fuel, it is still necessary, for properly determining the most commercially economical measure of expansion, to consider the other reasons governing it. On line 23 of Table No. 2 will be found expressed comparatively, the capacity of cylinder required with the different measures of expansion to develop, *ceteris paribus*, equal useful powers; and it will be seen that these capacities increase very rapidly with the measure of expansion. Now, as the cost, weight, and space occupied by engines of the same type are sensibly in the ratio of the capacities of their cylinder, the quantities on line 23 must enter largely into the commercial determination of the question. The adoption of a smaller capacity affects it as follows: There will be saved a part of the first cost proportional to the difference of capacity; and, also, a larger proportion of the annual repairs which are a greater per centum of the first cost of large engines than of small ones. And as engines are short-lived, as well as very costly, there is saved, too, the sinking fund necessary to cover the difference of the two costs. Of course, there is still to be added the difference of annual interest on these differences. The smaller engines will occupy a proportionally less space in the vessel and have a correspondingly less weight. They will, also, from their greater simplicity of construction, fewer and smaller parts, and greater proportional strength, be much less liable to break-downs, and be much quicker and easier repaired, and in many cases with the means on shipboard, when, with larger engines, the vessel would have to be taken into port. There will consequently be avoided much more of those enormous losses incident to such delays. The simplicity alone of the valve-gear, when no independent expansion-valve, or cut-off contrivance is employed, is sufficient to counterbalance a slight adverse difference in the economic effect of the fuel.

A difference in the economy of the fuel will affect the choice as follows: There is the difference in the annual cost of coal, and in the first cost and after repairs of the boiler for equal powers, which will be larger as the economy is less. There is also the difference between the space occupied by the boiler and coal, and between their weights, both of which will be greater in the ratio of the less economy of the fuel. As in the case of other expenditures there is the difference of sinking fund and of interest on the greater first cost of the boiler.

The quantities, then, that govern the choice of the most commercially economical measure of expansion, are those on lines 22 and 23 of Table No. 2. An examination of these quantities shows, that, in rapport of fuel alone, the most economical measures of expansion are those corresponding to cutting off the steam at $\frac{7}{10}$ths, and at $\frac{4}{9}$ths of the stroke of the piston from the commencement, the difference between the two being the very slight one of only 2 per centum of the greater quantity, while to obtain this the size of the engine must be 18 per centum greater. Of course, with this wide disparity, the choice, in every point of view, is decidedly for cutting off at the $\frac{7}{10}$ths. An estimate, however roughly approximative, made on the preceding considerations, and for the weights, bulks, and cost of average practice, will show the advantage to be so great as to conclusively answer the question.

For all measures of expansion less than that due to cutting off at $\frac{4}{9}$ths of the stroke of the piston, the loss, both in the economical effect of the fuel, and in the increased size of engine, rapidly increases, until, when cutting off at $\frac{1}{8}$th of the stroke, the loss in fuel alone is 19 per centum of the cost of the power when cutting off at $\frac{7}{10}$ths, while the size of the engine must be 127 per centum greater. If the point of cutting off be lessened to $\frac{4}{45}$ths of the stroke, the loss of economy in fuel alone reaches the enormous amount of 44 per centum of the cost of the power when cutting off at $\frac{7}{10}$ths; while the size of the engine must be 241 per centum greater.

As the experiments with the different measures of expansion were made with equal initial cylinder pressure, it is necessary for that condition to be maintained in order that the foregoing comparison of their relative economy shall be rigorously exact. But there is another and common case under which it is very desirable to also know the relative economy in rapport of fuel of different measures of expansion, namely, that of the same engine functioning with the same velocity of piston and doing the same useful work in equal time; that is, making stroke for stroke of piston with the same load. This case involves not only the effect of different measures of expansion *per se*, but of different initial cylinder pressures; for it is obvious that, under these conditions, the initial cylinder pressure must be made sufficiently higher when using larger measures of expansion to furnish the requisite equality of net pressure on the piston.

The principal economic effect of maintaining the high initial cylinder pressure, is to make the differential of the total pressure and the sum of the back and friction pressures more. This effect would, of course, be eliminated when comparing results from total pressure alone, as that pressure includes the back and friction pressures; now, if we would have the same piston develop equal useful power in equal time, that is, make revolution for revolution with the same load, we must have the net pressure equal, and as the total pressure is the sum of the net, friction, and back pressures, and as the last two are constant, we shall have the same total pressure to produce the same net pressure; consequently the numbers which express the comparative economic efficiency of the different measures of expansion in rapport of total power, will also express their comparative economic efficiency in rapport of net power when the initial cylinder pressure is adapted to give the same net pressure with the different measures of expansion. Hence the quantities on line 21 of

Table No. 2, express the comparative economic efficiency of the different measures of expansion in rapport of fuel, when the same engine is employed with the same velocity of piston and net pressure upon it. This deduction is rigorously exact, if we except the modification due to the difference of temperature in the cylinder during a stroke of the piston; but the correction for this, whatever may be its amount, will be in favor of the less measures of expansion, because, as with them, in order to obtain the same net pressure on the piston, the initial cylinder pressure, and consequent temperature of the entering steam, will be much less, while the temperature of the back pressure remains constant, the difference will be much decreased, and the condensation due to these extremes proportionally lessened. The quantities on line 21 of Table No. 1 may, therefore, be accepted as exact experimental determinations for the conditions in question, subject to the modification, just explained, due to the difference of the steam temperatures. Referring to this line, we find but slight differences in the economic effect obtained with the various measures of expansion between those due to cutting off the steam at $\frac{4}{45}$ths and at $\frac{7}{10}$ths of the stroke of the piston from the commencement, both inclusive. The extreme difference, within this range, is only $8\frac{1}{8}$ per centum of the least economical result, which is that given by the shortest point of cutting off. The most economical result is obtained when cutting off at $\frac{3}{8}$ths the stroke, and it is more economical than the result due to cutting off at $\frac{7}{10}$ths the stroke, by $4\frac{3}{4}$ per centum of the latter. The next most economical result is obtained when cutting off at $\frac{1}{4}$th the stroke, and it is more economical than the result due to cutting off at $\frac{7}{10}$ths the stroke, by $3\frac{3}{8}$ths per centum of the latter. The economic results due to cutting off at $\frac{3}{10}$ths and at $\frac{1}{8}$th of the stroke, are sensibly the same, and are greater than the result obtained when cutting off at $\frac{7}{10}$ths of the stroke by only $\frac{3}{4}$ths of one per centum of the latter. The economy of the $\frac{4}{45}$ths point of cutting off is less than that of the $\frac{7}{10}$ths point by $3\frac{3}{8}$ths per centum of the latter. From this comparison it will be perceived that, even under the most advantageous conditions for the greater measures of expansion, the lesser measures give equality of economical result in rapport of fuel. If the proper corrections could be made for the difference of cylinder temperature due to the different measures of expansion, it would doubtless be found that the economic result obtained when cutting off at $\frac{7}{10}$ths of the stroke of the piston is not exceeded when cutting off at any less fraction.

Finally, as steam-engines are required at different times to develop different powers, it may be asked whether the graduation of their power below the maximum cannot be effected more economically by increasing the measure of expansion and preserving the same initial cylinder pressure, than by reducing the initial cylinder pressure, and preserving the same measure of expansion. The former method involves the employment of an expensive and complicated adjustible expansion gear; the latter requires only a simple throttle-valve. Now, as all the experimental quantities required for the determination of this problem are to be found in Table No. 2, we will anticipate the question and reply to it here.

In the following table, the quantities on line 1 are the same as those on line 18 of Table No. 2. They are the total pressures in pounds per square inch, exerted on the piston during the different experiments. The quantities on line 2 are the sum of those on lines 4 and 6 of Table No. 2. They are the sum of the back and friction pressures in pounds per square inch of piston. The quantities on line 3 are the net pressures on the piston in pounds per square inch. They are the remainders of the quantities on line 1, after subtracting those on line 2.

The quantities on line 4 are the per centum which the quantities on line 3 are of those on line 1. They show the proportion which the net pressure is of the total, or, in other words, the proportion utilized of the total pressure. These quantities represent the relative net power that would be obtained per unit of

weight of fuel, using the steam with the same measure of expansion, but with the different net pressures on line 3; and by dividing unity by each of them, we obtain a new set of quantities that show the relative cost in fuel of the unit of net power. Now, taking as unity the quantity (84·6) in the column headed $\frac{7}{10}$, we obtain the quantities on line 5, and they represent, relatively, the cost in fuel of a net horse power when it is obtained by simply changing the initial cylinder pressure, without changing the measure of expansion, to the pressures required for giving the net pressures on line 3.

The quantities on line 6 are those on line 22 of Table No. 2, but arranged for the quantity in the column headed $\frac{7}{10}$ as unity instead of that in the column headed $\frac{11}{12}$. They represent, comparatively, the cost in fuel of a net horse power due to the net pressures on line 3 obtained from the same initial cylinder pressure, by simply changing the measure of expansion to those corresponding with the fractions of the stroke, in the headings of the columns, at which the steam is cut off.

Starting from the net horses power developed by the engine when cutting off the steam at $\frac{7}{10}$ths of the stroke of the piston, and graduating that power to those developed when cutting off at the other fractions in the heading of the table, the per centum of loss or gain in weight of fuel, per net horse power, which would result from making that graduation by changing the initial cylinder pressure with the throttle, or otherwise, instead of retaining this pressure and making the graduation by changing the measure of expansion with an adjustable cut-off, is shown on line 7.

| NUMBER OF LINE. | FRACTION OF THE STROKE OF PISTON COMPLETED WHEN THE STEAM WAS CUT OFF. | | | | | | |
|---|---|---|---|---|---|---|---|
| | $\frac{11}{12}$ | $\frac{7}{10}$ | $\frac{4}{9}$ | $\frac{3}{10}$ | $\frac{1}{4}$ | $\frac{1}{6}$ | $\frac{4}{45}$ |
| 1 | 34·0 | 31·1 | 27·1 | 22·9 | 20·1 | 16·4 | 12·5 |
| 2 | 4·8 | 4·8 | 4·8 | 4·8 | 4·8 | 4·8 | 4·8 |
| 3 | 29·2 | 26·3 | 22·3 | 18·1 | 15·3 | 11·6 | 7·7 |
| 4 | 85·9 | 84·6 | 82·3 | 79·0 | 76·1 | 70·7 | 61·6 |
| 5 | 0·985 | 1·000 | 1·027 | 1·061 | 1·111 | 1·196 | 1·373 |
| 6 | 1·116 | 1·000 | 0·978 | 1·059 | 1·073 | 1·188 | 1·442 |
| 7 | Gain, 13·1 | . . | Loss, 4·9 | Loss, 0·2 | Loss, 3·8 | Loss, 0·8 | Gain, 6·9 |

From line 7 of the above table, it will be perceived that if the net power have to be reduced from that which is obtained from the net pressures of 26·3 pounds per square inch of piston, with the steam cut off at $\frac{7}{10}$ths of the stroke of piston, to that which would be obtained from a net pressure of 22·3 pounds, it will be 4·9 per centum more economical to effect it by preserving the same initial cylinder pressure, and increasing the measure of expansion to that which is due to cutting off at $\frac{4}{9}$ths of the stroke, than to effect it by simply reducing the initial cylinder pressure by throttling without changing the measure of expansion.

If it be required to still further reduce the power to that which would be obtained by the net pressure of 18·1 pounds per square inch of piston, it will be one-fifth of one per centum more economical to effect it by preserving the original initial cylinder pressure and increasing the measure of expansion to that due to

cutting off at $\frac{3}{10}$ths of the stroke, than to effect it by reducing the initial cylinder pressure by throttling, retaining the measure of expansion due to cutting off at $\frac{7}{10}$ths of the stroke. And so on throughout the table.

From the foregoing it will be perceived that if we have an engine with the steam-valve set to cut off at the fixed point of $\frac{7}{10}$ths of the stroke for developing a maximum power, graduations to the degree of less power most favorable for being effected by an adjustable cut-off, may be made by the throttle alone, with a loss of less than five per centum of the fuel which would have been required had such graduations been effected by increasing the measure of expansion with the adjustable cut-off and retaining the maximum initial-cylinder pressure. When the reduction to be made in the power is very great, there is an absolute gain by making it with the throttle instead of with the adjustable cut-off. These results are exclusive of the less condensation produced by the less extremes of temperature in the cylinder when the reduction of power is made by the throttle, and which will be against the economy of the adjustable cut-off. In fact, the two modes of reducing the power may be considered equal in rapport of economy of fuel; but in every other respect the choice is immeasurably in favor of the throttle-valve.

The experiments have been made under the most favorable conditions, not only for obtaining exact comparative results, but for obtaining the maximum for the large measures of expansion. The boiler pressure was sufficiently high; the back and friction pressures the minimum in practice; the organs of the engine were well proportioned, and functioned properly; the boilers furnished easily an abundant supply of steam, and never gave the slighest evidence of foaming. Every circumstance combined to render the results even hypercritically unexceptionable; yet, in all respects, and under all possible changes of condition, they are conclusive against the popular belief in favor of high measures of expansion; for cutting off the steam at $\frac{7}{10}$ths of the stroke of the piston is scarcely recognised as working it expansively.

The results obtained from this engine are rigorously applicable to all others in which saturated steam is employed in a cylinder not steam-jacketed, and show conclusively the utter futility of attempting to realize an economical gain in fuel under such conditions by expanding the steam beyond the very moderate limit of one-and-a-half time; and that if the expansion be carried to three times, a positive loss is incurred. Also, that if measures of expansion as high as those due to cutting off the steam at $\frac{1}{6}$th or $\frac{4}{45}$ths of the stroke of the piston are employed, the economy is considerably less than with steam used absolutely without expansion.

The important fact, too, must not be overlooked, that all the claims for great economical gain by the use of steam very expansively have been asserted for engines of the same type as that employed for the experiments, and which are the kind almost universally employed for river and marine purposes—the engines in which the cylinder is steam-jacketed, or in which superheated steam has been attempted, consisting of but a few exceptional cases. There is nothing, however, in the discordance between the general impression of the economy of expansion and the results of our experiments that should prevent confidence in them; for it is only one more illustration of the well-known fact that the histories of all sciences are but records of mistakes and misconceptions arising from the application of fallacious theories, which once plausibly advanced, were long believed in from an unwillingness to investigate for ourselves, but which exploded at the first touch of the *experimentum crucis.*

With regard to steam-jacketed cylinders, there are no published experiments on the results of which we can safely base an opinion. Such a set of experiments have yet to be made, and are of the first consequence in engineering; but judging from our personal experience, we are by no means sanguine that, under proper conditions, with engines of and above the medium size, any gain greater than a few per

centum can be made; or that this gain will not be as great when the steam is used without expansion as when it is used expansively. The results of the present experiments are an additional admonition of the extreme unsafety of depending on inference in the physical sciences, and teach that only experimental determinations, under proper conditions, can be relied on for the solution of such problems.

In reporting these experiments we have given not only all the observed data and their calculated results *in extenso*, but also an accurate narrative of the manner by which this data was obtained, and of the method in which the calculations were made, in order to put the reader in such thorough possession of the whole case that he may completely conceive all the accompanying conditions.

We have, moreover, in the discussion of these experiments, given not only the direct and indirect results we arrived at, but we have stated, too, the acknowledged physical laws on which they were based, and described the inductive processes by which they are derived, in order that the reader may himself judge both of the propriety of the one, and the soundness of the other.

All of which is submitted, with the highest respect, by,

Sir,

Your obedient servants,

B. F. ISHERWOOD,
THEO. ZELLER,
ROBERT H. LONG,
ALBAN C. STIMERS,
*Chief Engineers United States Navy.*

HON. ISAAC TOUCEY,
*Secretary of the Navy.*

## ADDITIONAL REMARKS IN CONNEXION WITH THE SUBJECT OF THE FOREGOING REPORT.

IT appears to the writer that there are a few points not touched in the foregoing Report which may with advantage be discussed in connexion with it. They relate chiefly to the reliability of the indicator measurement of power, when using the steam expansively; the theoretical dynamic value of steam used without expansion; the causes of the condensation of steam in the cylinder; and the effects of steam-jacketing and steam superheating.

### OF THE INDICATOR.

The instrument habitually employed by engineers for the measurement of the power developed by steam-engines, is the Indicator of WATT. From the ease with which it can be applied and managed, and from the great practical difficulties attending the use of all other means of measurement with large powers, it may indeed be regarded as the only one practicable, and its errors, whatever they may be, are far less than those of any other that could be substituted. Further, its determinations are so various and so complete, when taken in connexion with other data, that no investigation could be made of the action of steam in an engine without its evidence.

When the same Indicator is employed in making comparative experiments with the same engine, its measurement of power, it would appear ought to be exact, ***comparatively***, at least, if not ***absolutely;*** and such would be the case were the steam used with the same velocity of piston, the same initial pressure and the same expansion; but when experimenting with different velocities of piston, initial pressures and degrees of expansion, even allowing perfection in the manufacture and adjustment of the instrument, there are some errors connected with its action which are impossible of elimination, and which become greater the more expansively the steam is used, the greater the velocity of the piston, and the higher the initial pressure.

Rigorously speaking, the sole thing ***directly*** measured by the indicator is the pressure in the cylinder, and that it determines for every point of the stroke of the piston. When the pressure does not vary during the stroke, as in the case of using the steam without expansion, its measurements are very nearly exact absolutely, and quite so comparatively; but if the pressure vary, as in the case of using the steam expansively, there will be an error both absolutely and comparatively, resulting from the fact that the movement of the indicator spring, owing to the inertia of its matter and of the matter of the piston, guide-rod, pencil and pencil holder connected with it, will be ***later*** than the change of pressure producing it. Now, as the steam-piston advances continuously, the ***difference of time*** between the change of pressure in the cylinder and the corresponding movement of the indicator-spring which measures it, must necessarily cause an error in the measurement of that pressure.

In the case of a continuously ***decreasing*** pressure, as in using steam expansively, the movement of the spring being always later than the decrease in the pressure, the expansion pressures as shown on the indicator diagram, will not be in their true places relatively to the position of the steam-piston; they will all be removed nearer the end of the diagram by the distance due to the difference of time and the velocity of the piston; the point of cutting off, too, will appear to be later than it really is; consequently the expansion pressures as measured on the diagram, will appear greater than they really were, and the error makes the power of the engine thus determined appear to be greater than it really was. Of course, ***ceteris***

*paribus*, the more expansively the steam is used the greater will be this error, and for two reasons. 1st. A larger portion of the diagram will be composed of expansion pressures. 2d. With equal initial pressures, the absolute variation of pressure, or space through which the indicator spring must descend, will be greater. The value of these errors cannot be estimated, but, even with the most perfect instrument, and under the most favorable conditions, they will amount to something, and that something, whatever it may be, will be in favor of the expanded steam, and more favorable, the more expansively the steam is used.

Under no circumstances can the variation in the pressure and the movement of the spring be simultaneous; the one, as a cause, must precede the other as an effect, for time is always required in the communication of motion to matter.

If to these considerations there be added the more or less unavoidable friction of the indicator piston and its guide-rod, tending to retard movement and make the time greater between the reduction of the pressure and the descent through the corresponding distance by the spring, we shall have no hesitation in admitting that, even with the best instrument, and under the most favorable conditions, the expansion curve must give an excess of pressure over the truth of probably several per centum.

Every engineer having an engine with a cut-off valve may easily test this fact for himself. Nothing more is necessary than when his engine is working with expanded steam at a uniform speed with a constant load to take a diagram, counting exactly the number of revolutions per minute at the moment; then immediately afterwards change to working the steam without expansion, and by means of the throttle bring his engine to precisely the same speed as before, and as soon as it becomes uniform, take another diagram; an exact measurement of the two will reveal the fact of different pressures, the highest being obtained with the expanded steam, though it is evident that the *real* pressures must have been exactly the same.

By indicator measurement of power, therefore, the expanded steam must always have an advantage over the non-expanded, due to the inherent action of the instrument itself.

As the whole error is an effect of *time*, it is plain that, *ceteris paribus*, it will be greater with greater velocity of piston; also that, with a spring whose index divisions are wide apart, it will be greater than with one whose corresponding divisions are closer together. The pressure at the end of the stroke of the piston will be exact in all cases, because the steam-piston, as it approaches the ends of its stroke, is gradually retarded by the crank till it stops.

If the same measure of expansion be used, then, *ceteris paribus*, the error will be greater as the initial pressure is higher; because the distance through which the spring will have to descend in equal time will be greater.

For the same reasons, the first part of the expansion curve will vary more from the true pressure than the last part; because, during the last part, both the velocity of the steam-piston and the difference of pressure will be less than during the first part. The first part of the expansion curve must always be considerably fuller than the truth.

Again, in the case of condensing engines, when the steam is expanded down below the atmospheric pressure, a great error in the indicator measurement of the power is caused by air leakage. All cylinders under such a condition leak air. It enters at the piston-rod stuffing-box, and through the relief water-valves, oil-cocks, indicator-attachment, joints of the heads, &c.; and from the moment the steam pressure within falls below the atmospheric pressure without, the steam becomes mixed with air, and the combined pressure of the two is indicated on the diagram. From this cause, too, the expansion-curve is much higher

from the atmospheric line to the end of the diagram, than it should be were there no air leakage, and after allowing for the effect of re-evaporation in the cylinder of the steam condensed during the first part of the stroke; and frequently at the end of the diagram where the speed of the piston is a minimum and the differential between the steam and atmospheric pressures a maximum, the discrepancy is enormous. Now all the air thus admitted, and which being shown on the diagram as pressure, is included in the calculation for the indicator power, adds really nothing to the power, as it has to be pumped out by the engine; but the additional calculated power thus obtained when the steam is used expansively, though purely fallacious, is, in experiments, credited to the expansion, and there is no way of eliminating it.

The same cause of air leakage also vitiates in great degree the calculations by indicator measurement of the weight of steam present in the cylinder at the end of the stroke of its piston, and gives rise to many *apparent* anomalies and discrepancies; further, in comparing the weight of steam discharged in a given time from the cylinder, as determined by the indicator pressure at the end of the stroke of its piston, with the weight evaporated in the boiler during the same time, in order to ascertain the condensation that has taken place in the cylinder, we shall obtain a result always *less* than the truth, and the less the more expansively the steam is used, especially if the cylinder initial pressure remains constant and the speed of the piston declines. It is of the highest importance in all experiments with condensing engines that there be the least possible air leakage; and in fact all accurate experiments on steam should be made with non-condensing engines, and with steam not expanded below the atmospheric pressure.

It must here be distinctly understood that, owing to the condensation of steam in the cylinder, the indicator is no measure, either directly or indirectly, of the weight of steam passed into the cylinder; consequently, even admitting it to be a correct meter of power, it can give no reasonable approximation of the cost of that power; hence any results based on the weight of steam consumed per indicated horse power, and calculated from the steam pressure in the cylinder at the end of the stroke of its piston, or at any portion of its stroke, will be wholly erroneous. The sole method of *accurately* determining the cost of the power is by measuring the weight of water pumped into the boiler, providing, of course, against loss by priming or leakage. The indicator does, indeed, if there be no air leakage, measure correctly the weight of steam in the cylinder at the end of the stroke of its piston, that is to say, the weight of that portion of the total steam that entered the cylinder which still retains its vaporous form, but it does not measure any of the portion of that total steam which has been condensed. Particular attention should be given to this fact, because nothing is more illusory than the determination by indicator of the weight of steam passed through the cylinder, and nothing is more common than to see results confidently announced by this determination.

We thus find that, by indicator measurement, the results both of power and of its cost, are greater than the truth, and greater the more expansively the steam is used. This is an important consideration to bear in mind when comparing the results of steam used with and without expansion, and with different measures of expansion.

## THEORETICAL DYNAMIC VALUE OF ONE POUND OF STEAM USED WITHOUT EXPANSION.

It will be satisfactory to know the theoretical dynamic effect, expressed in horse power, of one pound of steam per hour of the total pressure, 34½ pounds per square inch, adopted in the experiments, and used without expansion.

According to FAIRBAIRN'S formula, the volume of steam of this pressure is 722 times that of the water

from which it is generated. A pound of water at 62° Fahr. has a bulk of 27·7274 cubic inches, and a cylinder of it 12 inches high would have a base of 2·31062 square inches.

Suppose a vertical cylinder of indefinite height and having a base of 2·31062 square inches, into which let a pound of water at 62° Fahr. be poured; and let a steam-tight piston, free to move without friction, be placed in contact with the water and loaded till it pressed the water with a total weight of 34½ pounds per square inch, that is, 19·8 pounds per square inch above the atmosphere. Let exactly heat enough be imparted to the water to vaporize it, and suppose none of this heat to be lost except by transmutation into work.

Under these conditions, if there were no condensation of steam by transmutation of its heat into work, the piston will ascend a height of $(722-1=)$ 721 feet, and lift with it a weight of $(2{\cdot}31062 \times 34{\cdot}5 =)$ 79·71639 pounds; but as 6¼ per centum of the steam would be condensed to furnish the heat transmuted into this work, the height to which the weight would be lifted is $(\overline{1{\cdot}000 - 0{\cdot}062} \times 721 =)$ 676·3 feet. Now, whatever may be the power required to raise a given weight a given height, the same power will be exerted by the same weight falling through the same height. Supposing, then, the steam to be instantly and totally condensed, the weight would fall, according to the laws of gravity, through the above height in $(\sqrt{676{\cdot}3 \div 16{\cdot}083} =)$ 6·485 seconds. The mean velocity was, therefore, $\left(\frac{676{\cdot}3}{6{\cdot}485} \times 60 =\right)$ 6257·22 feet per minute, and the power exerted by the falling weight was $\left(\frac{79{\cdot}71639 \times 6257{\cdot}22}{33000} =\right)$ 15·115 horses. This power was exerted, however, only during 6·485 seconds instead of one hour. Now, if one pound of water gave 15·115 horses power, one horse power will be obtained for $\left(\frac{1{\cdot}000}{15{\cdot}115} =\right)$ 0·06616 pound of water per 6·485 seconds, and during one hour, or 3600 seconds, one horse power will require

$(6{\cdot}485 : 0{\cdot}06616 :: 3600 :)$ 36·727 pounds of water.

As there is no appreciable condensation in a well cased cylinder when the steam is used without expansion, exclusive of that which is caused by the development of the power, the above weight of water increased by the loss in the clearance and steam passages will be the practical quantity required per *total* indicated horse power per hour. The loss by clearance and steam passages may be taken, when the steam is used without expansion, at one-sixteenth of the volume of the cylinder; increasing the 36·727 pounds by this addition, we have 39 pounds of water per total indicated horse power. The total indicated horse power, it will be remembered, includes the back pressure from the uncondensed vapor in the cylinder.

If we estimate the back pressure at 3·5 pounds per square inch of piston, then, as the total pressure is 34·5 pounds, the gross effective indicated horse power will be obtained for $(\overline{34{\cdot}5 - 3{\cdot}5} : 39 :: 34{\cdot}5 :)$ 41·4 pounds of water. If the evaporative efficiency of the boiler be 10 pounds of water per pound of coal from the temperature of the feed-water, the gross effective indicated horse power will be obtained for 4·14 pounds of coal per hour with surface condensation, and with jet condensers, at sea, for about one-eighth more, or 4⅝ pounds of coal.

The proportion of the steam that, under the foregoing theoretical conditions, would be condensed to furnish the heat transmuted into the power of lifting the weight is easily obtained as follows, premising that the thermal equivalent of 772 foot-pounds is one pound of water heated one degree on Fahrenheit's scale.

The weight was 79·71639 pounds, and as it was raised to the height of 676·3 feet, the power developed was $(79{\cdot}71639 \times 676{\cdot}3 =)$ 53912·194557 foot-pounds, which being divided by 772 becomes 69·834449, the number of pounds of water that heated one degree will express the thermal equivalent of the power. Now,

taking the total heat of steam of 34·5 pounds pressure per square inch at 1192·3° above the zero of Fahr., and the temperature of the water at 62° Fahr., the above thermal equivalent is equal to vaporizing $\left(\frac{69\cdot834449}{1192\cdot3-62}=\right)$ 0·062 pound of this water, and as the weight of steam producing the power was one pound, the proportion condensed is obviously $6\frac{1}{4}$ per centum.

## CAUSES OF THE CONDENSATION OF STEAM IN THE CYLINDER.

If the change in the cylinder from steam to back pressure vapor could be made *without the deposition and re-evaporation of dew*, the temperature of the interior metallic surfaces would be but very little affected, because vapors, like permanent gases, receive heat with difficulty either by contact or by radiation, and because of the small specific gravity of the back pressure vapor. The only heat that could, in this case, be carried off from the cylinder surfaces to the condenser, would be that which was abstracted from the metal in superheating, rigorously, the back pressure vapor. None of the heat radiated from the interior surfaces of the cylinder, not absorbed by the vapor, would be lost, for, the cylinder being a complete enclosure, it would be intercepted by the opposing surface. The loss of heat due to the change of temperature in the metal of the cylinder, divides into two cases, namely, one when the steam is used without expansion, the other when it is used expansively. When the steam is used expansively we suppose its pressure and temperature to remain constant up to the point of cutting off, after which both fall continuously to the end of the stroke of the piston. Now, supposing there be no condensation from the expansion of the steam *per se*, from the heat transmuted into the power developed by the engine, or from external refrigeration, what is there to cause any greater deposition of dew on the interior surfaces of the cylinder when the steam is used expansively than when it is used without expansion? Under these conditions, when it is used without expansion the only cause of condensation has been seen to be the slight loss of heat due to the superheating of the back pressure vapor as a gas. The experiments of REGNAULT have determined that the *total* heat of steam increases with its pressure; consequently, when the same weight of steam is expanded, *i. e.* reduced from a higher to a lower pressure, there is less heat required to maintain it in the vaporous form than before, and instead of being condensed it will be superheated; such, indeed, is the deduction made by REGNAULT in his celebrated memoir. This, of course, supposes the steam to be unaffected by the containing vessel, or by molecular change in itself; and that what is called the total heat of steam is really a homogeneous and indivisible quantity producing but one effect.

If the temperature of the metal of the cylinder be higher than that of the steam expanding on it, this steam can absorb heat from the metal only after the manner of a permanent gas, and the absorption would go to still further superheat the expanded steam. The abstraction of heat from the metal of the cylinder during a stroke of the piston, due to this cause, we have seen, must be very slight indeed, and, of course, the condensation of steam during the next stroke, to re-supply it, will be proportionally small. It thus appears that the quantity of heat absorbed from the metal of the cylinder by the steam, during its expansion, is only the very little it can take up as a permanent gas. Little as that is, however, it makes the heat absorbed from the cylinder greater when using the steam expansively than when it is used without expansion; because the abstraction of heat from the interior surfaces, due to the superheating of the back pressure vapor, will be the same in both, and there will be, additionally, the abstraction due to the superheating of the expanding steam. This is the only difference between the two cases: the cause of the abstraction of heat is precisely the same in both, and is, whether the steam be used with or without expan-

sion, the superheating of steam or vapor acquired as a gas from the higher temperature of a metallic envelope.

There remains to be noticed, however, that, in the case of using the steam expansively, whatever superheating the expanding steam may obtain at the expense of the temperature of the metal of the cylinder, goes to increase its dynamic effect, and is, therefore, to that extent, no loss; while whatever superheating is expended on the back pressure vapor is a total loss. On the whole there is probably but a trifling difference in the loss by the lowering of the temperature of the cylinder metal by the superheating of the steam or vapor within, whether the steam be used with or without expansion; and in either case the loss is too insignificant for a practical estimate.

I have been thus prolix on this point, because it is the general, though, as we have seen, erroneous impression that the enormous difference in the quantity of condensation within the cylinder, when the steam is used with and without expansion, is due purely to the greater refrigeration caused by the reducing temperature of the expanding steam, and it is important to show how very small must be the difference due to this cause, if, indeed, it be at all sensible, and how enormously it must be exaggerated to have attributed to it even a small portion of the immense discrepancy experimentally ascertained to be the fact.

In addition to the cause just discussed, the others producing condensation in the cylinder are the development of power by the engine, external refrigeration, the expansion of the steam *per se*, and the reduction of the temperature of the metal of the interior surfaces of the cylinder by the re-evaporation of the water deposited on them in the form of dew by the aggregate condensation produced by all the foregoing causes.

As regards the condensation of steam by the transmutation of heat into the power developed by the engine, it is plain that as the quantity thus condensed does not vary greatly from the direct proportion of the power developed, it will be so nearly proportional to the total weight of water evaporated, whether the steam be used with or without expansion, that no difference of practical consequence can result in the two cases. Further, it is nearly certain that the water of condensation due to this cause is not deposited at all upon the cylinder surfaces, but remains suspended amid the steam like water bladders, or as fog or cloud is suspended in the air. For the condensation must take place throughout the entire mass of the steam as would happen by sending a chill through it. If the water of condensation, due to this cause, should not reach the cylinder surfaces, as it is nearly certain it does not, it can have no effect in producing any further loss by its re-evaporation from those surfaces. Consequently, in comparing the different condensations when using steam with and without expansion, or with different measures of expansion, the comparison should properly be confined to the quantities that remain after omission of what is due to the production of the power.

The condensation of steam in a cylinder, due to external refrigeration, is practically insignificant; it probably does not exceed one per centum of the water evaporated in the boiler; but be it what it may, it evidently is a less absolute quantity the more expansively the steam is used, supposing the cylinder initial pressure to be the same; because, in a given cylinder, this refrigeration is a function of difference of internal and external temperature solely, and that difference is obviously less the more expansively the steam is used. There are no means of ascertaining what, under these conditions, would be the condensation due to external refrigeration *proportionally* to the weight of water evaporated in the boiler, but, probably, the ratio would not vary practically in the two cases.

When the steam is used with and without expansion, and with initial pressures not equal but proper for

giving the same mean effective pressure throughout the stroke of piston, both the positive and relative condensation due to the external refrigeration will be sensibly equal.

We thus perceive that, as regards the production of condensation in the cylinder by external refrigeration, the quantity, which is but small in any event, will be sensibly in the same proportion to the weight of water evaporated in the boiler, whether the steam be used with or without expansion, and, consequently, cannot be one of the causes of the enormous difference in the cylinder condensation found to exist in these cases.

Now, if but little water of condensation can be deposited upon the cylinder surfaces from the causes thus far discussed, there can, of course, be but correspondingly little refrigeration effected in the metal by its re-evaporation, and this little will, as we have seen, be sensibly the same whether the steam be used with or without expansion; consequently, we must seek some other cause for the production of the enormous difference known by experiment to exist in these two cases. As we have already discussed all the *external* causes that can be productive of condensation in the cylinder, we must look to some *internal* cause, some special molecular change in the steam when used expansively, and which does not obtain when it is used without expansion.

Let us examine what would happen if water were converted into steam in vacuo. In this case, there being no external pressure upon it, the bulk of the steam would be defined when the force of the attraction of its watery particles for each other would just balance the repulsive force of the heat. In this condition the steam could have no temperature or sensible heat, for the entire quantity expended in its vaporization would be latent. Further, during such vaporization or enlargement of bulk no *external* work would have been performed. Now, as heat cannot be expended without producing its mechanical equivalent, and as, under the hypothesis, no external work has been done, it follows that the whole latent heat has been employed in performing *internal* work on the watery atoms, which work is measured by their removal apart from the distance they possessed in the state of water to the distance they are found to have in the state of steam. The latent heat has been expended in giving this much motion to the watery atoms against the force of their attraction. Hence we find the latent heat of steam to be simply the measure of the internal work performed on its watery atoms; in fact, it is the heat which has been transmuted into the mechanical work of the separation of those atoms, and which has, therefore, really ceased to exist. It is no reply to say that, by returning steam to the condition of water, we can re-develop or make sensible its entire latent heat, for that would only be a re-conversion of mechanical work into the heat which produced it. We can have the same element in the form of mechanical work, as evidenced in the enlargement of the bulk of water to its bulk in the state of steam, or in the form of sensible heat capable of producing physical effects on other matter, but we cannot have it in both forms at once. In this view of the subject, the quantity of latent heat in steam should vary in some ratio of the distance to which its watery atoms are removed asunder. It should be less with a less separation and more with a greater separation. In other words, the latent heat of steam should vary in some inverse ratio to its density, and this is known experimentally to be the fact.

Let us suppose that the steam has been generated in vacuo and has the bulk due to that condition; also that it has no temperature or sensible heat, but that its entire heat is latent; and suppose, further, that this steam is surrounded with an envelope which suffers it to neither lose nor gain heat from external temperature. Now, let this envelope be, by external power, gradually contracted to diminish the space within. From the instant the contraction commences, the steam acquires temperature or sensible heat and density, and continues to acquire them until the space is reduced to that which was originally occupied by the water

from which the steam was generated. In this condition the steam will have the density of water, but it will still be steam, and its entire heat will be sensible, for with its atoms at their original distance apart it can have no latent heat. Whence comes the sensible heat? It is derived solely from the transmutation of the external power producing the compression of the steam and is its thermal equivalent. As the sensible heat thus generated is in addition to the heat of vaporization, and is imparted to the steam as a gas, it may be remarked that all steam under pressure will be truly superheated, the superheating being a consequence of the compression and measured by the sensible temperature. The only rigorously saturated steam is that which exists under no pressure.

Now, let us suppose a reversal of the contracting process of the envelope, and let it be gradually enlarged by external power. In this case, the steam will expand but without performing external work; and in expanding its latent heat will continually increase at the expense of the sensible heat which had resulted from the compression: but by the decrease in the pressure, a less quantity of sensible heat will be required to satisfy the condition of equilibrium, consequently, there might be heat enough absorbed from the sensible heat to furnish the increase in the latent heat and still leave sensible heat enough to satisfy the condition of equilibrium with the reduced pressure, in which event no condensation would follow the expansion of the steam. But the heat absorbed from the sensible heat by the increasing latent heat in the expanding steam may be so great as not to leave sensible heat enough to maintain the condition of equilibrium with the reduced pressure, in which event a condensation will follow of such a portion of the steam as will liberate heat enough to maintain the remaining portion in the vaporous form. This would be the case of the condensation of steam by its expansion *per se*. And whether expanding steam is thus affected or not, depends wholly on the fact, determinable by experiment only, whether the latent heat increases faster than the sensible heat diminishes. The problem has been accurately solved by REGNAULT, and his researches show that the latent heat does increase faster than the sensible heat diminishes, and, consequently, there must result in all cases of expanding steam a condensation due wholly to the expansion *per se* and quite irrespective of the production of external power.

From our point of view it will be perceived that not only is the change of state undergone by water in its conversion into steam accompanied or effected by the conversion of heat from the sensible into what is called the latent state, but that after being in the vaporous form—and still preserving it—any molecular change, as expansion or compression, is caused or accompanied, respectively, either by change of sensible into latent heat, or by development of sensible heat—not from latent heat but by transmutation of power. In both cases the sensible heat disappearing or reappearing is the thermal equivalent of work done either by the steam internally upon its own molecules, or by an external power upon the steam, as in compression: consequently there is no such thing as latent *heat*, that term representing in fact a certain amount of *mechanical work* performed by the steam upon its own molecules, and due to, and measured by the conversion into it of a certain quantity of sensible heat whose disappearance has been described as its becoming "latent," when, in fact, this disappearance should have been accounted for by its annihilation as heat; its equivalent appearing in the form of mechanical work. The only means we have of determining heat is the thermometer, what does not affect that instrument is not heat, and when heat is consumed without producing an effect upon the thermometer, we must search for the mechanical work into which it has been converted. It is therefore erroneous to say that the total heat of steam is the sum of its latent and sensible heats, it is the sensible heat alone; what is called the latent heat being the mechanical equivalent of a certain quantity of sensible heat which has disappeared in the production of molecular change, but which will reapppear by the reversal of that change, the mechanical work being then con-

verted back into sensible heat. In all these operations we see transmutation and transfer but nowhere loss of any of the elemental force to whose different manifestations the terms heat, mechanical work, &c., are given. Hence, too, no practical error results from considering what is called "latent heat" as heat—the same as sensible heat—inasmuch as, though not heat *per se*, it is an exact measure and equivalent of it. The one in computation, can be correctly substituted for the other, but the two are ***in esse*** different.

In the case of generating steam of different pressure in two vessels by the application of the same flame during the same time, the quantities of steam that would be generated will, by no means, contain equal quantities of heat. In the steam of higher pressure, two effects have been produced, namely, a certain amount of mechanical work (latent heat) has been done upon the water by the flame, and a certain amount of heat (sensible heat) imparted. The latter alone is available for work. In the steam of lower pressure, also, the flame has produced the same two effects but in different proportions; it has done a greater amount of mechanical work upon the water (measured by greater latent heat) and imparted less (sensible) heat; but in both cases the sum of the work done and of the heat imparted will be equal; and as the work is simply transmuted heat, this sum can, in both cases, be correctly expressed in terms of heat alone.

Now, calling the total heat of steam the sum of the latent and sensible heats, it is evident that whether the total heat be constant for all densities, or whether it increases or lessens with the density, depends entirely on the fact of the proportion between the diminution of the latent and the increase of the sensible heat. If the latent heat diminishes faster than the sensible increases, the total heat would decrease with increase of density; and, if, on the contrary, the sensible heat increase faster than the latent diminish, the total heat will increase with increase of density; and, finally, if the latent heat diminish as the sensible increase, the total heat will be constant for all densities. Which of these hypotheses is true has been ascertained in the only way possible, namely, by the direct experiments of REGNAULT; and from them it is well known that the sensible heat increases with increase of density faster than the latent heat diminishes, and, consequently, that the total heat of a given weight of steam is greater with greater densities or pressures. Conversely, with decrease of density the latent heat diminishes faster than the sensible increases. Therefore, when steam is reduced by expansion from one density to a less, the latent heat or work done internally upon its molecules increases, and to the extent of that increase annihilates an equivalent of the sensible heat, and this equivalent being more than the difference of the sensible heats under the two pressures, there does not remain sensible heat enough to maintain the whole weight of steam in the vaporous form under the new conditions of equilibrium: part of the steam will, therefore, condense till enough of the mechanical work of its vaporization be re-converted into sensible heat to satisfy these new conditions. It is manifestly impossible to annihilate a quantity of heat in steam by transmuting it into mechanical work without condensing so much of the steam as will supply the thermal equivalent, and it matters not, in this view, whether the work done be internal or external.

Let the reader now carefully observe that what is called the latent heat of steam is really no heat at all, but a term representing the mechanical work done internally in effecting molecular change in the watery atoms; that a certain quantity of heat rendered latent means just that quantity annihilated as such and converted into work; that this latent heat ***increases*** with *decrease* of density, *faster* than the sensible heat supplying it ***decreases*** normally to the new conditions of equilibrium; and that a condensation of such a portion of the steam results that the mechanical work of the vaporization of this portion will be sufficient when reconverted into sensible heat to adjust the equilibrium under the different proportions of latent and sensible heats normal to the reduced pressure. This is what is called by the writer the condensation of

steam by expansion *per se;* it is due wholly to work performed internally on its molecules; it is irrespective of external work, and is additional to any liquifaction due to such. Now this condensation must take place equally throughout the mass of the steam, like the condensation due to the power developed by the engine, and the resulting watery spray, instead of being deposited like dew upon the interior surfaces of the cylinder, may, if not too great, continue suspended, fog-like, in the steam, and pass with it into the condenser. With high measures of expansion, there will probably be more of this spray deposited upon the cylinder surfaces than with low measures, because, there being a greater quantity of it and the steam having less density, it will be more difficult to hold in suspension.

There remains to notice the condensation of steam in the cylinder, due to the cooling down of its interior surfaces by the re-evaporation of the water deposited upon them in the form of dew by the aggregate condensation produced by all the foregoing causes. This condensation divides into two cases, namely, when the steam is used without expansion, and when it is used expansively.

If we suppose the steam to be used without expansion, and that, during a stroke of the piston, the interior surfaces of the cylinder become covered with a fine dew from the condensation of a portion of it due to any or all of the causes just discussed except that of expansion *per se;* and that this dew and the metal upon which it rests have the temperature of the boiling point of water normal to the pressure in the cylinder during the stroke; then, when the piston has reached the end of its stroke and the exhaust-port opens, the steam pressure will fall to that of the back pressure vapor, and the dew will evaporate until its temperature and that of the metal upon which it rests fall to the temperature of the boiling point of water normal to the back pressure. The vapor thus formed flows to the condenser through the open exhaust-port, and carries with it the heat of vaporization, which, having been absorbed from the metal, must, consequently, be re-supplied during the next stroke by the condensation of enough steam to furnish it; and so on, continually. By this alternate process we may suppose the interior surfaces of the cylinder, to a certain depth, to be cooled down, during each stroke of the piston, from the temperature of the steam to that of the back pressure vapor, and to be re-heated through the same number of degrees. The heat required for this re-heating is, of course, a total loss.

If the steam be used expansively, the cylinder pressure begins to fall from the moment the expansion-valve closes, instead of at the end of the stroke of the piston when the exhaust-port opens, and as the re-evaporation of any water of condensation begins with the reduction of the pressure, which is its efficient cause, and as the pressure continues to reduce from the closing of the expansion-valve to the end of the stroke, the re-evaporation will also continue during this time. As the steam thus re-evaporated, however, produces a dynamic effect upon the piston, it does not to this extent operate a loss. At the end of the stroke of the piston, the exhaust-port opens, and a further reduction of pressure takes place with a further re-evaporation; but as all the steam re-evaporated in the cylinder while the exhaust-port is open, flows into the condenser without having produced any dynamic effect on the piston, the heat contained in it is a total loss. But, if re-evaporation begins from the closing of the expansion-valve and continues to the end of the stroke of the piston, condensation also accompanies it between the same limits, because, as the recession of the piston continues to uncover fresh portions of the cool cylinder, the hotter steam expanding upon them must, in consequence, continue to undergo some condensation. Thus, during the whole time that the steam is expanding, the cylinder is simultaneously a boiler and a condenser. During the time from the commencement of the stroke of the piston to the closing of the expansion-valve, it is a condenser; and from the opening of the exhaust-port to the commencement of the stroke of the piston, it is a boiler.

Starting, now, with the same initial cylinder pressure and temperature, it is plain that the more expan-

sively the steam is used, the greater will be the time during which the cylinder surfaces will be exposed to steam of lower temperature, but it does not follow that there will be any more reduction of the temperature of the metal from that cause, than when using the steam without expansion. That temperature can only be reduced by external radiation, and by the loss of heat carried into the condenser through the medium of the vapor, and we have seen that the loss by superheating the expanding steam and back pressure vapor, is not only very small but sensibly equal, whether the steam be used with or without expansion, consequently any great reduction can only be effected by first causing a considerable deposition of dew and then re-evaporating it, and thus transferring the heat of its vaporization from the metal of the cylinder to the condenser. Now, as we have seen that with the exception of the condensation of steam produced by its expansion *per se*, there is nothing to cause more deposition of dew when using the steam expansively than when using it without expansion; and with equality of condensation there would, of course, be equality of loss by re-evaporation; but as it is proved experimentally that there is a much greater deposition of dew when using the steam expansively, and the greater the more expansively it is used, there follows that the deposited dew must be principally the result of the expansion *per se* of the steam. Hence we have, due to using steam expansively, not only the loss in the condensation by its expansion *per se*, but the additional loss due to the heat transferred from the cylinder to the condenser in the re-evaporation of the water of that condensation.

When steam is used without expansion, the condensation in the cylinder, exclusive of that which is due to the production of the power, probably does not exceed one or two per centum; but when it is used very expansively the condensation, *in excess of the re-evaporation*, rises to the enormous proportion of 30 and 40 per centum of the steam evaporated in the boiler, and, of course, when the re-evaporation, which in any case must be something, is added, these figures will be increased. Most of this condensation, however, I believe to be due to the expansion *per se*, particularly with moderate measures of expansion, and not to the effect of re-evaporation, which becomes great only with large measures of expansion.

The foregoing discussion, of course, excludes the condensation due to the cooling effect that would be produced by the evaporation of any of the boiler water that might be projected upon the cylinder surfaces by priming; and of the water condensed in the steam-pipe by external refrigeration and driven by the steam current upon the cylinder surfaces. The more expansively the steam is used, however, the less will be the probability of loss from either of these causes.

As we can neither add to nor modify the properties of matter in order to adapt it for our special purposes, we find that, in subjecting it to a change of condition (which is simply making an experiment on the resources of nature), we do, not only what we intended to do, but a great deal more. Of this general law, which has rendered worthless many an ingenious invention in the industrial arts, the expansive use of steam is a striking particular case, for the great additional power that should be derived from it is so nearly neutralized by the excessive condensation which accompanies it, that but a very narrow margin of economic gain remains.

## STEAM JACKETING AND STEAM SUPERHEATING.

Having thus discussed the causes of condensation in the cylinder, it is proper to consider how their action may be modified by "steam-jacketing" or by "superheating" the steam, both of which, as regards the saving of heat in the cylinder, can, it will be shown, be economical only to the extent they may be able to prevent the deposition of the water of condensation on the interior surfaces of the cylinder, and, consequently, the loss of heat transferred from the metal of the cylinder to the condenser in its re-evaporation.

Now, how would these effects be modified by steam-jacketing? Steam-jacketing is enveloping the cylinder with steam of a higher temperature—little or much—than that within it; and if we suppose the heat from the jacket-steam to be transmitted through the metal of the cylinder quickly enough to replace the heat as fast as it is lost by the steam within the cylinder, all deposition of water of condensation on its interior surfaces will be prevented, and, of course, the loss of heat transferred from the metal of the cylinder *to the condenser* by the re-evaporation of this water will be saved.

Admitting that the effective action of all the causes of condensation, except the re-evaporation of the dew deposited on the interior surfaces of the cylinder, cannot be lessened by steam-jacketing, but that it can, by furnishing enough heat to counteract their refrigerating effects, maintain in the vaporous form, *during the entire stroke of the piston*, all the steam entering the cylinder, we shall have prevented condensation, and thereby obtained the full dynamic effect of the steam, but by an expenditure of heat equivalent to the refrigerating influences. This is for the steam side of the piston, now for the exhaust side: when the exhaust-port opens, there being no water of condensation on the interior surfaces of the cylinder, no heat can be transferred by its re-evaporation from the steam in the jacket to the condenser. Consequently, with the steam-jacket, the only loss of heat that would be caused by the free communication of the cylinder with the condenser, would be the superheating imparted by the jacket to the back pressure vapor in the cylinder as a gas; for, as each stroke of the piston pushes a cylinder full of this vapor into the condenser, whatever heat has been bestowed upon it is a total loss.

When the steam-jacket is not employed, the steam being used expansively and accompanied by deposition of dew, there will be, as already explained, a re-evaporation of this dew from the instant the expansion-valve closes to the end of the stroke of the piston, and resulting from the lessening pressure whereby the temperature of this dew and the metal on which it rests are maintained above the boiling point normal to the reducing pressures. The heat of this re-evaporation, derived partly from the dew itself and mainly from the metal of the cylinder, is not a total loss, because the steam generated by it exerts a dynamic effect upon the piston. Part of it, however, will be a loss, because as a certain time must elapse between the condensation and the re-evaporation, and as, during that time, the piston is in motion, the dynamic effect that would have been produced by the re-evaporated steam during that time is lost. The steam-jacket, then, under these conditions, produces a saving on the steam side of the piston, measured by the loss of dynamic effect of the re-evaporated steam during the time it remained water in the cylinder.

We thus perceive, that, supposing the steam-jacket to act efficiently, that is to say, to supply heat as fast as needed to counteract the refrigerating influences within the cylinder, and thereby prevent condensation, it would obtain on the steam side of the piston when the steam is used expansively, a greater economic effect from the steam, and on the exhaust side it would save all the heat that is transferred from the metal of the cylinder to the condenser, *while the exhaust-port is open*, by the vaporization of whatever water of condensation remains in the cylinder at the expiration of the stroke of the piston.

In general, there are certain refrigerating influences acting within the cylinder, which can be counteracted only by the expenditure of an equivalent of heat, and the sole reason why it is more economical to supply this heat from a steam-jacket than from the metal of the cylinder is that, in the former case, the supply is furnished on the steam side of the piston only (neglecting the slight expenditure and loss in superheating the back pressure vapor on the exhaust side), and is therefore all utilized dynamically; for on the exhaust side, as there will be, as a consequence, no water of condensation to absorb heat in its re-evaporation, there can be no supply furnished. In the latter case, the supply is furnished not only on the steam

side of the piston but on the exhaust side too, because on both sides there is water of condensation to be re-evaporated, and all the heat furnished on the exhaust side is a total loss, because the steam generated by it produces no dynamic effect upon the piston. Of the heat furnished on the steam side of the piston, part is lost, because there is a part of the stroke during which the steam generated by that heat remained in the form of water.

When the steam is used without expansion, there will be no re-evaporation during the stroke of the piston on its steam side, consequently no heat will be absorbed either from steam-jacket or metal of cylinder to supply it. On the exhaust side the jacket will save, as in the case of using steam expansively, all the heat that would pass into the condenser from the metal of the cylinder by the vaporization of water of condensation during the time the exhaust-port remained open. The gain by the jacket, therefore, will be less the less expansively the steam is used, and a minimum when the steam is used without expansion.

Such appear to be the general laws governing the economy of the steam-jacket, but there are some modifications to be made to the result due to them, *per se*, by other results produced by it.

In the first place, the loss of heat by external radiation is greater with the jacket than without, because the temperature of the steam within the jacket is higher than the mean temperature of the steam in the cylinder, and because it has a greater external surface. As a corollary, with equal initial cylinder pressure this loss will be less the less expansively the steam is used, and a minimum when the steam is used without expansion. Also, this loss will be less the less the steam is throttled.

In the next place, more heat is lost with the jacket than without, in superheating the back pressure vapor as a gas, because a higher temperature is imparted. And, further, when the engine is employed during short periods only, as for instance one-half of each twenty-four hours, there is the heat lost in bringing up the temperature of the metal of the jacket to that of the steam at each start. This loss will be measured by the weight of the jacket and the specific heat of its metal. In a money point of view, too, there is the cost of the jacket and its appendages; and in a vessel, additionally, the weight and space occupied.

On the other hand, with a condensing engine, the jacket causes a saving of the power required to be expended on the air-pump in pumping out the water of condensation formed in the cylinder when the jacket is not used, and the injection water required to condense the vapor generated by the re-evaporation in the cylinder of this water. Also, there will be less power expended on the feed-pump, because, for equal powers developed by the engine, less feed water will be required.

From the foregoing discussion it will be seen that the utility of the steam-jacket is confined to its action in *preventing* deposition of dew on the interior surfaces of the cylinder, and the saving it effects will be confined exclusively to the heat of evaporating such dew not utilized in dynamic effect on the piston. It cannot save any of the loss of heat due to other refrigerating causes in the cylinder, such as external radiation, superheating the back pressure vapor, annihilation due to the production of the power, or to the expansion *per se* of the steam, if the steam be used expansively. If, from any cause, the steam-jacket should not *entirely* prevent the deposition of dew, its economy will be limited to the amount it does prevent. In the Cornish, which is a single-acting engine, it acts much more efficiently than in the ordinary double-acting engine, because there is *double time* for the heat of the jacket steam to penetrate the metal of the cylinder.

It will be understood that the alternate heating and cooling of the metal of the cylinder does not extend through it, but only to a certain and, in general, extremely slight depth from the interior surface, even at a maximum. When the cylinder is in use this depth is constantly fluctuating according to the thermal and refrigerating influences within. Beyond this depth the temperature of the metal will not be *at all* affected

by these variable conditions; for when the heat-wave due to the entering steam has advanced to this point it will be made to recede by the refrigerating influences. The alternations of temperature could not extend through the metal unless its transmission of heat was instantaneous instead of being slow and continuous. What, then, regulates the temperature of the remaining thickness of the metal and of the outside surfaces of the cylinder? Simply the temperature of the atmosphere and the conduction of heat from the valve-chest through that portion of the metal which is outside of the fluctuating heat-wave caused by the variable thermal conditions within the cylinder. Thus the temperature of the outside surfaces of the cylinder will remain constant as long as the temperature of the steam in the valve-chest and of the atmosphere remain constant.

As the value of the steam-jacket depends on its efficiency in preventing deposition of water of condensation on the interior surfaces of the cylinder, and as its economy is measured by the amount of that prevention, it is of the highest importance to ascertain 1st. The quantity of this deposition, and 2d. How much of it is prevented by the jacket.

In the case of using steam without expansion, as there is no re-evaporation on the steam side of the piston and no condensation by expansion *per se*, a comparison between the weight of feed-water pumped into the boiler and the weight of steam exhausted at the end of the stroke of the piston shows so little discrepancy, after deducting the condensation due to the production of the power, as to demonstrate how very slight is the aggregate loss by external radiation, superheating the back pressure vapor and vaporization of deposited dew; one or two, or three per centum at furthest, will cover it. It is not probable, as before stated, that the water of condensation due to the production of the power is precipitated on the cylinder surfaces; on the contrary, it doubtless remains suspended in the mass of the steam as very fine mist, and is swept out by the exhaust. Although there must be some deposition of dew even when using the steam without expansion, yet it is so small that practically the quantity may be considered as nothing, and, of course, a steam-jacket in this case would be a completely useless incumbrance.

As the steam, however, is used more and more expansively, the discrepancy, after deduction of the condensation due to the production of the power, becomes greater and greater between the weight of feed-water pumped into the boiler and the weight of steam exhausted from the cylinder at the end of the stroke of the piston into the condenser, until, when very high measures of expansion are employed, as, for instance, cutting off at one-eleventh of the stroke, it rises to nearly forty per centum of the feed-water. In this extreme case, if the steam and exhaust-valves are such as to prevent egress of water, the slapping of the piston at the end of each stroke shows the presence of water within, and it thus appears that the metal of the cylinder has not been able to furnish sufficient heat for the re-evaporation of the quantity of water of condensation deposited. When this is the case the temperature of the interior surfaces of the cylinder must be reduced very nearly to that of the boiling point of water under the pressure of the back pressure vapor.

What could cause this great difference in the quantity of internal condensation when using the steam with and without expansion? Is it to be supposed that, in the latter case, when the cylinder was filled with back pressure vapor half the time and with steam the other half, and yet no appreciable condensation followed; that, in the former case, when the cylinder was only exposed additionally on the steam side of the piston to a steam pressure lower indeed than that of the initial steam but higher than that of the back pressure vapor, there should have been caused thereby the enormous condensation of forty per centum? The only difference in the two cases is that, when the steam is used without expansion, the cylinder is exposed half the time to the back pressure vapor and half the time to the steam, while, when it is used expansively, it is exposed half the time to the same back pressure vapor, and part of the other half to the

initial steam pressure, and the remainder of the time to the less pressure of the expanding steam. Now, in both cases, if we except condensation by expansion *per se*, there was only the difference of heat carried off from the metal of the cylinder by the superheating, in one, of the back pressure vapor, and by the superheating, in the other, of the same back pressure vapor, and, additionally, of the expanding steam; and as we have seen that, in the first case, the loss of heat from this cause was practically nothing, it could, in the second case, have been but very little, if at all, appreciable; we are, therefore, driven to the conclusion that the enormous difference of condensation experimentally known to exist, is due to the expansion of the steam *per se*, partly directly and partly indirectly from the loss of heat by re-evaporation of the water precipitated by it. Granting that the condensation which takes place by expansion *per se* with extreme high measures of expansion approaches forty per centum, it will be easily understood that the mass of steam, reduced in density by the expansion, cannot sustain in suspension so great a load of water, and that a large part of it must be precipitated upon the cylinder surfaces, where its re-evaporation by the heat of the metal causes an additional loss of heat and equivalent condensation. The forty per centum here mentioned as experimentally known, is exclusive of the steam re-evaporated during the expansion part of the stroke of the piston, as this steam is included in the indicator measurement; consequently, as there must be some re-evaporation, and may be a great deal, the real condensation may much exceed the forty per centum, it may be fifty or sixty per centum; but let it be what it may, it is impossible to separate it into what belongs directly to the expansion *per se*, and what to the re-evaporation which is an indirect result of that expansion.

With less measures of expansion there is less condensation, but, in the experiments, there does not appear to hold any ratio between the increase of the condensation and the increase in the measure of expansion. It will be recollected, however, that the condensation, as experimentally determined, is only the difference between the tank and the indicator measurements, the first being the weight of steam passed into the cylinder, and the last the weight of steam passed out of it, and that in the last the steam re-evaporated during the expansion part of the stroke of the piston is included; could this vitiating quantity, and the air leakage into the cylinder after the expanding steam had fallen below the atmospheric pressure, be eliminated, a regular progressive increase of condensation would doubtless be found to accompany a corresponding regular increase in the measure of expansion.

It appears, then, that when the steam is used expansively, there is a loss due to re-evaporation from the interior surfaces of the cylinder, which might be prevented by the steam-jacket, did it act with theoretical efficiency, that is to say, did it impart heat fast enough to the expanding steam to prevent the deposition of water by overcoming the refrigerating influence as fast as it arose. The amount of this loss, and the consequent saving in preventing it, cannot, as we have seen, be estimated; but I am of opinion that it is not very great, even in extreme cases. The jacket, however, cannot act with anything approaching theoretical efficiency, because the transmission of heat through the metal of the cylinder will be very slow on account of its thickness, imperfect conductibility, and the short time in which the changes of temperature are to be effected. The jacket, then, can only partially prevent deposition of water, and, of course, can only to the extent of that prevention effect a saving. In actual practice, with double-acting engines, the measures of expansion habitually used, and the thickness given to the cast iron of the cylinder, I am of opinion that no important gain will be obtained by the employment of a steam-jacket. This opinion is given as the general result of my experience and inquiry, but I hold it subject to correction by the result of experiment when such shall have been properly made. The question is an important one, and well worth an experimental

determination by any government. The cost is insignificant compared with the benefits to be derived from a settlement of the issue, let it eventuate how it will. The experiments should embrace a complete set with different measures of expansion, made in a similar manner to those on board the U. S. Steamer "MICHIGAN," and both with and without the jacket.

SUPERHEATED STEAM may be used as a substitute for the steam-jacket, and its economic effect ***as regards the steam in the cylinder*** will be the same, namely, limited to the saving due to the ***prevention*** of the deposition of the water of condensation. For this purpose the steam must have such an excess of superheating when entering the cylinder that the temperature of saturation will not be reached till it leaves the cylinder.

Now, as the refrigerating influences are greater the more expansively the steam is used, it is obvious that the steam on entering the cylinder should be given a greater excess of superheated temperature as the measure of expansion increases. The proper excess of temperature to be given above that of saturation will, therefore, vary according as the steam is used more or less expansively. When the steam is used without expansion, the superheating temperature and the economic effect fall to zero.

The heat for superheating the steam must, of course, come from the fuel; and if we suppose the boiler correctly designed with such proportions that the products of combustion on entering the uptake to the chimney have only the proper temperature for producing the draught, it is plain that all the heat abstracted will be at the expense of the efficiency of the boiler by reducing its draught; besides, with such a boiler the temperature of the uptake (the superheating apparatus is supposed to be located in it) will be so low that the steam cannot obtain sufficient heat to maintain it in the vaporous form throughout the stroke of the piston, especially with high measures of expansion. If, on the contrary, the boiler be so proportioned that the products of combustion are delivered into the uptake at a temperature that will give the steam an efficient superheating, a loss instead of a gain will follow, the reduced evaporative effect of the fuel being greater than the increased effect of the superheating. It has been abundantly proven that it is far more economical in the production of power to expend a given quantity of heat in the generation of steam from water than in the superheating of that steam out of contact with water.

The advantageous use of superheated steam appears to be confined to the case of bad boilers, that is to say, of boilers which deliver their products of combustion into the chimney uptake at temperatures above 600° Fahr., and which prime badly. The higher the uptake temperature, the more the priming and the larger the suaface and capacity of the superheating pipes, the greater will be the economy; for under these conditions, the superheating tubes become in effect an enlargement of the boiler and tend to convert bad proportions of heating surface and steam room into good ones. The water primed over from the boiler is evaporated in the superheating tubes, a large part of whose surface is thus made to heat water as well as steam. The water primed over is not only converted into steam by the waste heat of the uptake, but the heat which had already been imparted to it in the boiler is saved, and its bad effects in the cylinder are avoided. The quantity, too, of water primed is diminished by the additional capacity of the superheating pipes. It is thus easy to perceive that, with the mal-proportions existing in many cases, a very desirable gain may be effected by superheating, while in other cases of correct proportions no advantage follows. On the whole, superheating appears to be an expedient whereby some portion may be recovered of the heat which should never have been lost; and whatever effect is produced in the manner above indicated, together with superheating proper, is a clear gain, for it has cost nothing as it has been obtained from heat that would otherwise have been wasted, and which was not needed for producing a draught.

It is possible, with sufficient excess of superheating temperature, to impart so much heat to the steam that the refrigerating influences will be counteracted to the end of the stroke of the piston and deposition of water on the cylinder surfaces prevented; a result which, with even moderate measures of expansion, cannot be commanded with the steam-jacket. Superheating, then, is both more efficient and more economical than the steam-jacket when the boiler proportions are such as to furnish enough *waste* heat to effect it. The influence of either on the expansive use, *per se*, of steam I regard as nothing. It will not in the least degree increase the economical result of using steam expansively, and the whole gain, *as regards the economy of the steam in the cylinder*, will depend entirely on the *prevention* of the loss of that heat which would be transferred to the condenser from the metal of the cylinder in the vaporization of the water of condensation that would otherwise be deposited upon it. When the superheating is done with a bad boiler there will be added, *as regards the economy of the fuel*, the result of whatever improvement in its proportions may be due to the superheating apparatus.

There are insuperable objections to the use of superheated steam even with bad boilers. It is not a practicable medium on account of the chemical changes it produces in the constitution of the metals with which it comes in contact. It rapidly oxidizes wrought iron; the tubes of the superheating apparatus, made of that metal, are soon destroyed by its use, and the valve-seats and interior surfaces of cast iron cylinders converted into plumbago; besides which there are the injurious effects due to the absence of the copious distilled water lubrication with the saturated steam.

The superheating apparatus is objectionable in itself on account of its weight, cost, space occupied, and inconvenient position; and particularly on account of its danger and the complicated system of valves thereby entailed; for provision must be made for shutting it off at a moment's notice and resorting to the use of saturated steam.

Attempts have frequently been made to use superheated steam, but they have never been attended with permanent success. A new scheme is patented, a trial made, great results announced by the interested parties, and in a few weeks the contrivance that was to have revolutionized the use of steam is found upon the scrap heap. In the machinery of art, as in that of nature, only the simplest forms are permanent.

## OF THE APPLICATION OF THE LAW OF MARIOTTE TO VAPORS AND GASES.

It is so far from being known that saturated steam expands according to the law of MARIOTTE, that it is certain it varies from this law, and, probably widely, even when raised by superheating to the state of a gas.

Aeriform substances are divided into vapors and gases; the division, however, is more popular than philosophic, for the only difference between them is in the relative force with which they resist condensation. Those that are condensible by a moderate diminution of temperature, the pressure remaining the same; or by a moderate increase of pressure, the temperature remaining the same, are called vapors; while the gases are those which require very great variations of these conditions before they will condense into liquids.

The gases are truly superheated vapors, that is, the boiling point of the liquid from which they are generated is, under ordinary pressures, so very low that ordinary temperatures are greatly higher than what is due to their pressure as saturated vapors; in other words, they are highly superheated, and can therefore be made to undergo great changes of compression and expansion without condensation. In this

view all the gases are condensible when the proper conditions of pressure and temperature can be commanded: and it is probable that each gas has its own conditions under which it will liquify.

When a compressing force is applied, the general law in regard to the elasticity of gases—known as the law of MARIOTTE—is that the compression is directly proportional to the compressing force; but experiment has shown that this law is not exact for any gas which has been experimented on, and for gases approaching the condition of a saturated vapor, as carbonic acid gas, the discrepancy between the theoretical law and the experimental result is very great, and with true vapors, as steam, the variation from the law is doubtless wider still.

The experiments of REGNAULT show that, in atmospheric air and some other incondensible gases, the rate of compression, though nearly proportional to the compressing force, is not exactly so, the compression increasing faster than the pressure; in hydrogen gas the opposite was the case. With carbonic acid gas—the most condensible of the gases—the rate of compression increased greatly faster than the pressure. It is to be regretted that no direct experiments on the rate of compression and dilitation of steam out of contact with water, comparatively with the compressing force, have been made. Enough is known, however, to render it certain that it varies widely from the law of MARIOTTE, even supposing no heat to be lost or imparted externally during the operation. Until the proportionality of the pressure of saturated steam to its volume enlarged by expansion out of contact of water, and neither receiving nor losing heat externally during the expansion, shall have been experimentally ascertained, we cannot determine how near the experimental results obtained from a steam-engine working steam expansively approach the *true* theoretical limits.

The experiments already made on different gases show that mere simplicity is no proof of a law in physics, and admonish us of the extreme danger of assuming physical truths analogically, arguing from one gas or vapor to another; they demonstrate that the only safe ground is careful and direct experiment in each case. The propensity of the human mind to generalize is very great—that is to say, to infer the universality of a fact found to obtain in a few cases; but no law can be properly extended to cover more than the experimental facts on which it is founded. There is not a single theory in science whose flank is not liable to be turned to-morrow by some fresh discovery which ought to harmonize with it, but is found to be in direct contradiction. The simplicity in question seems only predicable of certain abstract conditions which we may, indeed, arbitrarily assign or imagine, but which have no foundation in nature and cannot be realized in matter as we find it.

The causes of the great discrepancy found to exist in steam-engines using steam with different measures of expansion, between the economy as promised by the law of MARIOTTE and as realized experimentally, may be summed up as follows, premising that the same initial and back pressures are supposed to be employed in the cylinder, namely:—

1st. The law of the expansion of steam is not rigorously that of MARIOTTE, even when condensation is prevented by superheating; the pressure decreases in a higher ratio than the volume increases.

2d. The condensation of steam in the cylinder due to the production of the power.

3d. The condensation of steam in the cylinder due to superheating the back pressure vapor as a gas.

4th. The condensation of steam in the cylinder due to its expansion *per se*.

5th. The condensation of steam in the cylinder due to external radiation.

6th. The condensation of steam in the cylinder due to the re-evaporation of water deposited on its interior surfaces.

7th. The loss of dynamic effect in the cylinder clearance and steam passages.

8th. The influence of the back pressure in the cylinder resisting the stroke of the piston.

9th. The influence of the pressure required to work the engine *per se.*

10th. The difference of dynamic effect due to an equal weight of steam used at the average cylinder pressure, and at the boiler pressure.

The mode of action of these causes, their value, and the conditions modifying them, will be found set forth in the Report of the Board of Naval Engineers on the Expansion Experiments at Erie; and in the preceding additional remarks in that connexion made by the writer.

With the exception of the 3d and 5th—both insignificant—all the above enumerated causes become rapidly more potent the more expansively the steam is used; and after a deliberate investigation of their value the unprejudiced reader will find it easy to understand how the gain by expansion is restricted to the very moderate limits of cutting off at seven-tenths of the stroke of the piston from the commencement; and which render inappreciable the effect of considerable variations in the measure of expansion, and of nice design and adjustment in the expansion-gear.

Such are the results of the much vaunted theory of expansion when it is submitted to the *experimentum crucis.* A theory which owes its universal acceptation to its extreme plausibility, the absence of properly conducted experiments, which, even when the services of qualified experimenters can be commanded, are too laborious and expensive to be lightly undertaken, and the incessant advocacy of those interested in cut-off patents.

The results of the expansion experiments made at Erie cannot be overthrown by argumentation. The only way in which they can be rebutted is by making an equally careful and extensive set of experiments under their conditions on another engine and obtaining contrary results.

The theory of expansion breaks down when experimentally tested, because it is fallaciously constructed. The true method of constructing a sound theory on any subject in physical science, is to commence by ascertaining the value of every quantity by direct measurement that appears in any way connected with it; and that, too, without inquiring, *a priori*, whether it will be useful for our purpose, for its utility cannot be known till the end. After we have accurately measured all the quantities, it is necessary to arrange them in tables, grouping them according as they appear related, in order to clearly obtain all the facts at one comprehensive view, and then, from these tables we can form general laws by gradually rising from particulars to generals. We must not seek for physical truths in metaphysical abstractions, but in the connexion subsisting among natural phenomena, and a theory inconsistent with a constant and uniform fact must be erroneous. What is sound theory? The *whole* of the knowledge we possess on any subject, put in such order and form that we can make a reliable practical application of it. Physical science is but a rational empiricism, that is to say, the results of the observation of facts confirmed, when they can be, by the operation of the intellect in showing them to harmonize with other facts; but owing to the extreme uncertainty of this operation and our limited knowledge, the observation of facts must not only furnish the basis for the theory, but their continued consistency must be relied on to sustain it, additional facts acting continually to re-shape or confirm theories.

In physical science an inquiry into causes is altogether vain and futile. HUME makes the keen observation that no copula had been detected between any cause and effect. We employ the language of causation because it is convenient, and gives precision to our ideas, but it is gratuitously applied to that which we know only as consecutive. Science has no concern but with the discovery of laws, and the laws of na-

ture (as they are called) are merely generalized facts; the facts must precede the laws, and we can only be certain of the truth of the law after we know *all* the facts relating to it. To arrive at results in this Baconian way, however, is a dry and laborious process. The *a priori* method of the Greeks is much more easy and entertaining. According to this system we commence with the laws instead of ending with them. We first, by drawing upon our stock of preconceptions, arrange what may be, ought to be, or must be, and, having thus determined the possible and impossible of the case, ignore all non-conforming facts and testimony with a fixed purpose to waste no time in their examination. The history of the expansion question is a complete illustration of this method. The whole theory is based upon a pure and simple abstraction, that of the idea of perfect elasticity in gases and vapors unaffected by any of the conditions of matter or of the steam-engine. And because it is so simple and so plausible, and apparently so conclusively proven by an inspection of indicator diagrams with the false confidence that they measure correctly both the power and its cost, that those interested in the manufacture and use of steam machinery, deriving their opinions from superficial books, cling to it with a prejudice amounting to fanaticism. Because they have not suspected and do not comprehend the causes operating to reduce the apparent gain, they will not believe in them even when their effects are practically proven. Now we are so far from knowing all the agents of nature and their various modes of action that it would not be good sense to deny any effect merely because in the actual state of our knowledge it happens to be inexplicable. But though we should be scrupulous in admitting it in such a case without a rigorous examination, yet when it has fairly borne that test, it should be accepted like any other physical fact, whether a satisfactory cause can be assigned or not. An unlimited skepticism is the result of a narrow mind which, reasoning on imperfect data, makes its own knowledge and extent of observation the standard and test of truth. It is, in short, mistaking the horizon for the bounds of the universe.

EVAPORATION GIVEN BY THE VERTICAL WATER-TUBE BOILERS

OF THE

# U. S. STEAMER "MICHIGAN,"

WITH THE

ORMSBY, AND BROOKFIELD COALS, AND WITH ANTHRACITE.

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It appeared to the writer that there might be appended with advantage to the Report made by the Board of U. S. Naval Engineers of the Experiments with the Machinery of the U. S. Steamer "MICHIGAN" to Determine the relative Economy of Using Steam with Different Measures of Expansion, a fuller exposition of the Evaporative Efficiency of the Coals used than appeared in that Report, and to accompany it with a description and analyses of them. This has been done in the present paper.

The anthracite was used for only one experiment, and solely in order to obtain data for comparing the evaporative power of the boiler with that of other boilers on the Atlantic coast in which anthracite is almost exclusively used.

The Ormsby coal, which was employed in all the experiments except two, is considered the best steam coal from the west of Allegheny Mountains coal-field, and will doubtless be extensively used on the northern lakes and the Ohio river. It is, therefore, of importance to a great steam navigation interest, and large section of country, to know its composition, characteristics, and comparative evaporative power.

The Brookfield coal is from the same locality as the Ormsby, and is identical with it.

An engraving and description of the boilers will be found in the Report before mentioned.

I have, accordingly, collected the data and results of all the experiments that were made with the "MICHIGAN'S" boilers, and combined them in the six columns of the table hereinafter given, being guided in the combination by the rate of combustion, placing those in one column whose rates did not differ enough to be of practical importance. In fact, the extremes varied but little from the mean.

## DESCRIPTION OF THE COALS.

ORMSBY COAL. The coal principally used in the experiments with the machinery of the U. S. Steamer "MICHIGAN," is known at Erie as the Ormsby coal. It is from the immediate neighborhood of Clarksville, Pennsylvania, about half-way between the cities of Erie and Pittsburgh, and about seven miles from the western State line of Pennsylvania. Clarksville is about eighty-five miles S. S. W. from Erie, and is on the northern border of the great west of Allegheny Mountains coal-field.

This coal, though containing a large proportion of volatile matter, is not bituminous, the gaseous matter, or the greater portion of it, appearing to be condensed in the pores of the fixed carbon in the free state, and not in the viscous condition of bitumen. It probably owes its low specific gravity of 1·24 to this fact, and to its small proportion of earthy matter, which averages only about 6 per centum when it is burned in the furnaces of a boiler.

In mechanical structure it is extremely friable, possessing but little cohesion and crumbling easily into fine dust. The hand is very much soiled by contact with it, a great quantity of soot-like powder adhering. The fracture is lamellar, and the planes of deposition are everywhere fissured at right angles by cross partings, which thus divide and sub-divide the masses into cuboidal fragments down to very small dimensions. The planes of deposition have a dull black color, while the cross partings exhibit a shining black appearance.

In the furnace the Ormsby coal kindles very easily and burns dry with great rapidity, even with a moderate draught, splitting up at once into minute cuboidal masses, and thus presenting an enormous surface for the emission of gas and the contact of air. These masses retain their form to the last, neither softening, agglutinating, nor swelling. When thrown into the furnace the coal quickly develops a large mass of very brilliant, yellowish, dense flame, which continues during the whole combustion, and is accompanied by considerable smoke, even under the favorable circumstances of the large combustion chamber and copious admission of air both to it and to the furnace in the case of the "MICHIGAN'S" boilers. Of course the smoke gradually becomes less as the combustion progresses, but it rarely ceased altogether from the "MICHIGAN'S" pipe. This smoke was of a brownish color. From the pipes of other steamers, however, using the Ormsby coal, and having no supply of air to the furnaces except through their grates, there continually rolled a full column of dense black smoke. The flame would sometimes burst out with slight explosions and in long white jets of intense brilliancy which would continue a considerable time.

This coal is probably exceeded by none in ease of ignition and rapidity of combustion; it requires very little attention or labor in the furnace, needing neither slicing nor stirring. It makes scarcely any clinker, even with the most intense combustion, and the ashes, which are very light, weighing only 25·6 pounds per cubic foot, fall freely through the grates. Heavy fires can be carried for twenty-four hours without requiring to be cleaned.

The weight of this coal in a merchantable state is 43·5 pounds per cubic foot, requiring 51·5 cubic feet in the bunkers to stow one ton of 2240 pounds.

The following are the proximate and organic analyses of the Ormsby coal, having the average proportion of refuse given during the experiments.

## PROXIMATE ANALYLSIS.

| | |
|---|---|
| Fixed Carbon, | 53·72 |
| Volatile matter exclusive of water, | 39·03 |
| Water expelled at 212° Fahr., | 1·25 |
| Ashes, | 6·00 |
| | 100·00 |

## ORGANIC ANALYSIS.

| | |
|---|---|
| Carbon, | 67·57 |
| Hydrogen, | 21·47 |
| Oxygen, | 2·67 |
| Nitrogen, | 1·04 |
| Water (mechanically present in the pores), | 1·25 |
| Ash, | 6·00 |
| | 100·00 |

Omitting the nitrogen, water, and ash, the composition is as follows, namely:—

| | |
|---|---|
| Carbon, | 73·68 |
| Hydrogen, | 23·41 |
| Oxygen, | 2·91 |
| | 100·00 |

One ton of 2240 pounds yields 8668 cubic feet of illuminating gas and 46 bushels of coke.

One hundred parts of the volatile matter, exclusive of water, has the following composition:—

| | Proportion of Constituent. | Ultimate Analysis of Constituents. | | | |
|---|---|---|---|---|---|
| | | Carbon. | Hydrogen. | Oxygen. | Nitrogen. |
| Olefiant Gas, | 6·50 | 0·92885 | 5·57115 | . . | . . |
| Light Carburetted Hydrogen Gas, | 39·23 | 29·42250 | 9·80750 | . . | . . |
| Carbonic Oxide, | 12·00 | 5·14820 | . . | 6·8568 | . . |
| Hydrogen, | 39·60 | . . | 39·60000 | . . | . . |
| Nitrogen, | 2·67 | . . | . . | . . | 2·67 |
| | 100·00 | 35·49455 | 54·97865 | 6·8568 | 2·67 |

Brookfield Coal. The Brookfield coal mine is situated only three miles distant from the Ormsby mine. The two coals are identical in appearance, action, mechanical and chemical constitution, and proportion of refuse. Its evaporative efficiency is doubtless equal to that of the other, though in the single experiment made with it it gave an inferior result by a few per centum.

Anthracite. The anthracite was the Lehigh from the north-eastern part of Pennsylvania, and possessed the well known characteristics of that coal. Its cohesion is great, requiring strong blows from a hammer to break moderate sized lumps. The fracture is semi-conchoidal, the color a brilliant jet, and

the lustre almost metallic. It scarcely soils the hand by contact. This coal ignites with difficulty and burns slowly and entirely by the surfaces, giving out during combustion an intense heat. The lumps neither soften, swell nor agglutinate. There is but little development of flame, and that only at the commencement of the combustion; the color is at first bluish and very faint, passing soon into a faint yellow, when the flame quickly disappears, leaving the incandescent fixed carbon alone. No smoke is at any time produced.

The anthracite contains a large portion of earthy matter that, vitrifying under the intense heat, forms large masses of clinker which require much time and labor to detach from the grates and withdraw from the furnaces. The furnace doors were consequently kept open longer than with the Ormsby coal, and the boiler was more cooled by the inrushing air. This alone, doubtless, considerably reduced the evaporative effect of this coal. The fires became rapidly dirty and needed much cleaning, and a great deal of slicing to keep up steam. In the single experiment made with the anthracite its combustion was forced to the utmost, and then failed to supply as much steam as was required.

The weight of the coal in a merchantable state, screened, the lumps being about the size of a goose's egg, was 54·5 pounds per cubic foot, requiring 41·1 cubic feet of bunker room to stow one ton of 2240 pounds. Specific gravity 1·55.

The following are the proximate and organic analyses of the anthracite, having the proportion of refuse given during the experiment.

## PROXIMATE ANALYSIS.

| | |
|---|---|
| Fixed carbon, | 72·00 |
| Volatile matter exclusive of water, | 6·97 |
| Water (mechanically present in the pores), | 2·50 |
| Ash, | 18·53 |
| | 100·00 |

## ORGANIC ANALYSIS.

| | |
|---|---|
| Carbon, | 73·41 |
| Hydrogen, | 2·23 |
| Oxygen, | 2·36 |
| Nitrogen, | 0·97 |
| Water (mechanically present in the pores), | 2·50 |
| Ash, | 18·53 |
| | 100·00 |

Omitting the nitrogen, water, and ash, the composition is as follows, namely:—

| | |
|---|---|
| Carbon, | 94·12 |
| Hydrogen, | 2·86 |
| Oxygen, | 3 02 |
| | 100·00 |

## TABLE—EXHIBITING

### STEAMER "

| NUMBER OF LINE. | |
|---|---|
| 1 | Duration of Experiment in hours, |
| 2 | Pounds of Water pumped into Boile |
| 3 | Pounds of Coal consumed, . |
| 4 | Pounds of Refuse from Coal, in Ash |
| 5 | Pounds of Combustible consumed, |
| 6 | Per Centum of Refuse from the Coa |
| 7 | Pounds of Coal consumed per hour p |
| 8 | Pounds of Combustible consumed pe |
| 9 | Pounds of Steam withdrawn from th |
| 10 | Number of times the Steam Room of t |
| 11 | Height of the Barometer, in inches |
| 12 | Steam Pressure in Boilers, in pounds |
| 13 | Temperature of the Water, in degre |
| 14 | Temperature of the Engine Room, i |
| 15 | Temperature of the External Atmos |
| 16 | Pounds of Water evaporated from te |
| 17 | Pounds of Water evaporated from te |
| 18 | Pounds of Water evaporated from to of Coal, . . |
| 19 | Pounds of Water evaporated from te of Coal, . . |
| 20 | Pounds of Water evaporated from te of Combustible, . |
| 21 | Pounds of Water evaporated from to of Combustible, . |

## MANNER OF MAKING THE EXPERIMENTS.

With these coals there were made the experiments recorded in the following Table.

The "*manner of making the experiments*" will be found detailed under that head in the preceding Report made by the Board of U. S. Naval Engineers of the experiments with the machinery of the U. S. Steamer "MICHIGAN" to determine the Relative Economy of using Steam with Different Measures of Expansion.

## EXPLANATION OF THE TABLE.

The following Table contains such of the data of the various experiments made with the machinery of the "MICHIGAN" at Erie, during the winter of 1860–61, as is necessary for the determination of the evaporative efficiency of the boilers with the three kinds of coal used and with the different rates of combustion employed. For facility of reference the columns are lettered and the lines numbered.

Columns A, B, C, and D contain the data and results with the Ormsby coal burned with different rates of combustion, arranged progressively, the lowest rate occupying column A, and the highest column D. Column E contains the only experiment made with the Brookfield coal. And column F the only one made with Anthracite.

The quantities in column A are the means of three experiments of seventy-two consecutive hours each, and of one of eighty consecutive hours. The quantities in column B are the means of four experiments of seventy-two consecutive hours each. The quantities in column C are the mean of three experiments of seventy-two consecutive hours, and one of thirty-six consecutive hours. The quantities in columns D, E, and F are for each column, the mean of seventy-two consecutive hours. In all these experiments the extremes differed but very slightly from the mean.

Line 1 contains the aggregate duration of the experiments for each column in hours.

Line 2 contains the total number of pounds of water evaporated from the temperature on line 13. The water, previously to being pumped into the boilers, was measured in a tank filled to precisely seventy cubic feet, and the weight is calculated from the number of tanks, and the weight of a cubic foot of water at 62° Fahr., namely, 62·321 pounds, corrected for the temperature on line 13, according to the elaborate researches of KOPP on the expansion of water.

Line 3 contains the total number of pounds of coal consumed during the time on line 1, weighed on a delicate platform scale of FAIRBANKS' manufacture. The coal was supplied to the furnaces evenly and with great regularity. The apertures in the furnace doors for the admission of air were open during the whole time; those beneath the bridge wall were opened whenever fresh coal was fired and allowed to remain open till it was well ignited, when they were closed.

Line 4 contains the total number of pounds of refuse in ashes, clinker, and dust withdrawn from the furnaces and ash-pits, and weighed *dry* on the same scales as the coal.

Line 5 contains the total number of pounds of combustible consumed. It is the remainder of the quantity on line 3 after deducting the quantity on line 4.

Line 6 contains the per centum which the quantity on line 4 is of the quantity on line 3.

The quantity on line 7 is obtained by dividing the quantity on line 3 by the number of hours on line 1, and the quotient again by 90—the number of square feet of grate surface in the two boilers.

The quantity on line 8 is obtained by dividing the quantity on line 5 by the number of hours on line 1 and the quotient again by 90.

The quantity on line 9 is obtained by dividing the quantity on line 2 by the number of minutes in the hours on line 1.

The quantity on line 10 is obtained by dividing the quantity on line 9 by 50·9—the number of pounds of steam at 35·15 pounds per square inch above zero, contained in the steam room of the two boilers and their steam chimney, and in the steam-pipe, cylinder, side-pipe, and valve-chest outside the valves. The pressure 35·15 pounds is the mean of all the experiments, from which the extremes differed but very slightly.

The quantities on lines 9 and 10 show the comparative velocity with which the steam is withdrawn from the boilers; but while the quantity on line 9 only shows it by the absolute weights withdrawn, that on column 10 shows it in comparison to the bulk withdrawn from. The boilers did not exhibit, during any of the experiments, the least sign of foaming or priming, as far as could be detected by the eye or ear, either in the glass water-gauge, gauge-cocks, or in the cylinder; but to whatever extent water may have been primed, or if it passed over in the vesicular state with the steam, the quantities on lines 9 and 10 would affect the apparent evaporation absolutely, because it would be measured in them; and comparatively, because more water would pass over as the steam room was more rapidly exhausted.

The quantity on line 11 is the height of the barometer, and is necessary as showing the density of the atmospheric air maintaining the combustion and affecting the velocity of the draught in the chimney; also, as showing the true pressure of the steam above zero.

Line 12 contains the pressure, by an ALLEN spring-gauge and by an open syphon mercurial-gauge, of the steam in the boilers in pounds per square inch, under which the water was evaporated. This quantity is necessary because the heat required to evaporate a given weight of water at a given temperature, varies with the pressure under which it is evaporated, being greater with greater pressures than with less. It is also necessary for determining the quantities on lines 16 and 17.

Lines 13, 14 and 15 contain respectively the temperature of the water, of the engine room, and of the external atmosphere in degrees Fahr.

Line 16 contains the quantities on line 2, corrected for the difference in evaporating the water from the temperature of 100° Fahr. and from the temperatures on line 13.

Line 17 contains the quantities on line 2, corrected for the difference in evaporating the water from the temperature of 212° Fahr. and from the temperatures on line 14.

In calculating the quantities on lines 16 and 17, the total heat of the steam of the pressure on line 12 is taken from REGNAULT's experiments.

Line 18 contains the quotients resulting from the division of the quantities on line 16 by the quantities on line 3.

Line 19 contains the quotients resulting from the division of the quantities on line 17 by the quantities on line 3.

Line 20 contains the quotients resulting from the division of the quantities on line 16 by the quantities on line 5.

Line 21 contains the quotients resulting from the division of the quantities on line 17 by the quantities on line 5.

The evaporation is calculated from the two temperatures of 100° and 212° Fahr. for facility of comparison with other experiments, as both are used for standards.

The experiments in column D with the Ormsby coal, and in column F with the anthracite, were made with the maximum combustion that the boilers would furnish.

## REMARKS ON THE RESULTS IN THE PRECEDING TABLE.

As the per centum of refuse in coal (line 6) is an accidental proportion, it is proper to take as the cost of evaporation in a philosophic—not a commercial—view, the weight of combustible consumed (line 5). Even this would require some modification in favor of the coal containing the larger per centum of refuse, because the action of the furnace changes that refuse from one state to another, and necessarily by an expenditure of a certain quantity of heat upon it which is thereby rendered not available for evaporative purposes. This refuse, when withdrawn from the furnace, has the temperature of the incandescent fuel that surrounded it, all of which is lost.

In remarking upon the results, the weight of water evaporated will be taken as that on line 16, which is the quantity that would have been evaporated by the coal consumed had the temperature of the water been uniformly 100° Fahr. instead of the different temperatures on line 13.

Dividing, then, the quantities on line 16 by those on line 5, we have, for the purposes of comparison, the quantities on line 20, which are the evaporation, in pounds of water, from the temperature of 100° Fahr. by one pound of combustible.

Observing, for the Ormsby coal, this evaporation in relation to the rate of combustion (line 8), it will be perceived that as this rate increases the evaporation decreases. The rate of combustion is in the following proportion, namely, 1·000, 1·775, 2·782 and 5·266. The corresponding evaporation is as follows, namely, 1·000, 0·970, 0·947 and 0·923; showing that, in rapport of economy of fuel, the heating surface was not too large in proportion to grate surface, even with the excessively slow combustion of 4·140 pounds of combustible, per hour per square foot of grate, and with copious admission of air both to the furnace and to the combustion chamber behind the bridge wall. If it be supposed that any water was primed over, or carried from the boiler with the steam in the vesicular state—and the velocity of the steam current from the boiler (line 10) was five times as great with the maximum as with the minimum rate of combustion—such water would go in direct ratio to apparently increase the evaporation; consequently, if any such transvection of water occurred, the above decreasing ratio for the evaporation will be too small, and the loss of economic effect due to increased rate of combustion will be greater than appears above.

In comparing the evaporative efficiency of the pound of combustible of the Ormsby with that of the pound of combustible of the Brookfield coal, it is proper to select the experiment with the former in which the rate of combustion approximates the closest with that of the latter. Column D contains this experiment, and by making the comparison it appears that the Brookfield coal was $\left(\frac{8{\cdot}325 - 7{\cdot}993 \times 100}{8{\cdot}325} = \right)$ 4 per centum inferior to the Ormsby. The proportion of refuse may be considered the same in both.

Comparing the evaporation of the Ormsby coal and the anthracite in the same manner, selecting the experiment in column C of the former on account of its furnishing the nearest approximation to equality in rate of combustion, it appears that the evaporative efficiency of the pound of combustible of the Ormsby coal is $\left(\frac{9{\cdot}088 - 8{\cdot}536 \times 100}{9{\cdot}088} = \right)$ 6 per centum inferior to that of the anthracite. But the proportion of refuse is very different in the two, being only 5·51 per centum with the Ormsby coal, while it rises to 18·53 per centum with the anthracite. With equal proportions of refuse, the evaporative efficiency of the

pound of combustible of the Ormsby coal would have a still greater inferiority than the above 6 per centum.

Allowing, as above determined, that the evaporative efficiency of the pound of combustible of the Brookfield coal is 4 per centum less than that of the Ormsby coal, it follows that it is 10 per centum less than that of the anthracite; and if proper allowance be made for the effect of the forced combustion of the anthracite, the greater proportion of its refuse, and the longer time the furnace doors were kept open, it will be more than 10 per centum less, probably as much as $\frac{1}{6}$th or $16\frac{2}{3}$ds per centum less.

The same allowance being made in comparing the anthracite with the Ormsby coal, the economic efficiency of the pound of combustible, *per se*, of the latter, will be not less than 10 per centum inferior to that of the former. Hence, if anthracite could be obtained with as small proportion of refuse as the Ormsby coal, it would, for equal weights, be 11 per centum more economical.

In engineering, it is very desirable to know the exact calorific value of one pound of carbon, that is to say, to know the number of pounds of water which one pound of carbon will heat one degree on Fahrenheit's scale, provided its combustion was perfect and the whole of the heat generated taken up by the water, the carbonic acid gas—the sole product of the combustion—passing off at the temperature of the air supporting that combustion.

I am aware that to this question the below mentioned authorities, of great reputation, have given the following answers, namely:

| | | |
|---|---|---|
| DULONG, | 13,118 | |
| DESPRETZ, | 14,241 | |
| LAVOISIER, | 13,723 | |
| FAVRE and SILBERMAN, | 14,083 | for diamond. |
| " " | 14,000 | for graphite. |
| " " | 14,544 | for wood charcoal. |
| ANDREWS, | 13,820 | " |
| Later by ANDREWS, | 14,186 | " |
| Mean of all, | 13,964 | |

Of these I consider the last three with charcoal the most reliable; the mean is 14,183. I believe that all of them, however, are too low, and for the following among other reasons.

The laboratory experiments by which the above results were obtained were made on a very small scale. The exterior surface of the apparatus was very large in proportion to the quantity of fuel consumed, and the loss of heat by radiation must, consequently, have been considerably more, proportionally, than is found in large boilers. It was impossible, too, to abstract the heat from the products of combustion down to the temperature of the external atmosphere. The perfection of the combustion was also uncertain. Finally, the charcoal experimented with contained a considerable per centum of moisture, which it holds with such tenacity that it cannot be completely expelled without exposure to a red heat.

The property possessed by charcoal of rapidly condensing large quantities of gases in its pores is well known; it absorbs vapors still more easily, and liquids most easily of all. Messrs. ALLEN and PEPYS determined experimentally the weight gained by recently ignited charcoal of different woods, after a week's exposure to the atmosphere, to be as follows, namely: Fir charcoal gained 13 per centum; Lignum vitæ charcoal 9·6 per centum; Box charcoal 14 per centum; Beech charcoal 16·3 per centum; Oak charcoal

16·5 per centum; and Mahogany charcoal 18 per centum. The absorption, which was of both moisture and air, was most rapid in the first twenty-four hours.

In the course of my professional life, I have frequently had to test the evaporative efficiency of boilers with Pennsylvania anthracite, making the tests with critical accuracy, carefully observing temperatures and all the attending circumstances, with correct determinations of every quantity; and I have found that, in the case of well proportioned vertical water-tube boilers, with air supply to the furnaces sufficient to ensure complete combustion but without being so much as to reduce the average temperature of the furnace below 2260° Fahr., and with heat absorbing surface sufficient to reduce the temperature of the products of combustion to about 375° Fahr.; that is, making the heat expended upon the products of combustion one-seventh of that generated in the furnaces when the temperature of the external air is 60° Fahr. $\left(\frac{2260-60}{375-60}\right)$ *the calorific value of one pound of the carbon, including the heat expended upon the products of combustion, was the heating of* 16,000 *pounds of water one degree Fahr.* This result is $\left(\frac{16000-14{,}183\times100}{14{\cdot}183}=\right)$ 12·8 per centum more than the mean obtained with wood charcoal by the chemists; but I am satisfied that it is very nearly correct. Allowing this to be the proper determination, the similar combustion of one pound of hydrogen should have the calorific value of 64,000 pounds of water raised one degree Fahr. An experimental determination by ANDREWS gave 60,854 pounds.

Let us apply these figures to the Ormsby and anthracite coals of the experiments in columns C and F of the table.

The composition by weight of the pure combustible of the two coals is as follows, namely:—

| | ORMSBY. | ANTHRACITE. |
|---|---|---|
| Carbon, | 75·89 | 97·05 |
| Hydrogen, | 24·11 | 2·95 |
| | 100·00 | 100·00 |

With these proportions, the calorific value of the pound of Ormsby combustible will be $(\cdot7589\times16000+\cdot2411\times64000=)$ 27,572 pounds of water raised one degree, and that of the anthracite will be $(\cdot9705\times16000+\cdot0295\times64000=)$17,416 pounds. Taking the anthracite's for unity, the evapotive effect of the combustible of the Ormsby coal should have been 1·583; whereas we have seen that the evaporative effect as realized, taking the anthracite's for unity, was for the Ormsby coal only 0·900. The anthracite was composed of nearly all carbon, and we may, for the moment, suppose it to have been wholly so, and that its evaporative effect was due to it alone. Then, as the carbon constituent of the Ormsby coal will, of course, give equal results per unit of weight, the discrepancy between the relative theoretical and practical evaporations of the two coals must be looked for in the hydrogen constituent. That constituent in the Ormsby coal was $(24{\cdot}11-2{\cdot}95=)$ 21·16 per centum greater than in the anthracite, and the inferiority of 10 per centum in the practical evaporation by the pure combustible of that coal proves the 21·16 per centum of hydrogen to have been only equal to $(21{\cdot}16-10{\cdot}00=)$ 11·16 per centum of carbon. In other words, the practical calorific value of the hydrogen element, as it existed in the coals, was sensibly one-half that of an equal weight of the carbon element, instead of being four times, making a difference of eight fold between the theoretical and practical results relatively with carbon.

In the case of these experiments, it will be observed that the boilers were proportioned purely for the hydrogen element. The admission of air was copious through perforations both in the furnace doors and

beneath the bridge walls. A large combustion chamber was provided between the bridge walls and tubes for the proper mingling of the gases with the atmospheric air, and for their retention in sufficient mass to preserve their temperature until chemical union was effected. And a large calorimeter was given to accommodate the large bulks of the resulting products of combustion. The Ormsby coal, too, was not a bituminous but a gaseous coal.

It will now be instructive to compare the practical with the theoretical evaporation given by the carbon. Assuming, as above determined, that the practical calorific value of the hydrogen is equal to one-half that of an equal weight of carbon, the pure combustible of the anthracite will be, referring to the organic analysis, $\left(73{\cdot}41 + \frac{2{\cdot}23}{2} =\right)$ 74·53 per centum of its weight; and as the pound of anthracite evaporated 7·404 pounds of water, its pure combustible evaporated $\left(\frac{7{\cdot}404}{{\cdot}7453} =\right)$ 9·934 pounds of water from the temperature of 100° Fahr. into steam of 19·5 pounds pressure per square inch above the atmosphere. Taking the total heat of steam of this pressure to be 1192° Fahr. and the calorific value of the pound of carbon to be 16,000 pounds of water raised 1° Fahr., we have, for its theoretical evaporation under the conditions, $\left(\frac{16000}{1192 - 100} =\right)$ 14·652 pounds of water. Hence, the practical result was

$$\left(\frac{14{\cdot}652 - 9{\cdot}934 \times 100}{14{\cdot}652} =\right) \text{ 32·2 per centum less than the theoretical.}$$

The mean temperature of the gases emerging from the tubes into the chimney uptake was 520° Fahr.; the temperature of the air supplying the combustion was 75° Fahr.; hence the temperature lost was (520 — 75 =) 445°, which, as the loss by the products of combustion was 32·2 per centum, would make the mean temperature of the furnace (32·2 : 445 : : 100·0 : 1382, and 1382 + 75 =) 1457° Fahr. With just air enough admitted to furnish the oxygen chemically required for the combustion of the fuel, the furnace temperature would be about 4500° Fahr.; consequently, in the present case, there had been admitted about three times as much air as was rigorously necessary.

The economic evaporation by the "MICHIGAN'S" boilers was below mediocrity, instead of being the maximum usually found with vertical water-tube boilers. It will be advantageous to ascertain the cause.

With boilers of the same type, the element exercising the greatest influence on the evaporative economy is the calorimeter, or cross area for the passage of the products of combustion between the heating surfaces. It exercises this influence as a proportion, referred both to the grate surface and to the heating surface. The calorimeter for maximum result with these boilers must be not only a certain proportion of the grate surface—not exceeding one-sixth nor less than one-eighth, but the absolute spaces between the tubes, crosswise the furnaces, should not exceed about 1⅜th inch for a length of tube box of about 7 feet. Now, in the boilers of the "MICHIGAN," the calorimeter was one-third—double the maximum—and the spaces between the tubes 3 inches for a length of tube box of 11 feet.

The effect of enlarging the calorimeter, *ceteris paribus*, is to admit more air to the furnaces in proportion to fuel consumed; the temperature of the furnaces is thereby proportionally reduced, and if we suppose the products of combustion to enter the chimney at a constant temperature, it is evident they will carry off with them a larger proportion of the total heat generated in the furnaces. Again, *ceteris paribus*, the effect of enlarging the calorimeter is to increase the bulk of gases enveloped by the heating surfaces, or, in other words, to decrease the heating surface in proportion to the heat to be absorbed, consequently, the products of combustion will enter the chimney at a higher temperature and carry with them from this cause a larger proportion of the total heat generated in the furnaces.

We thus see that, *ceteris paribus*, enlarging the calorimeter beyond the proportion determined by practice for the maximum result, reduces the economic evaporation by increasing the temperature of the escaping products of combustion, both absolutely, and relatively to the furnace temperature. It effects the former by operating a virtual decrease of the heat absorbing surface, and the latter by reducing the temperature of the furnace.

With vertical water-tube boilers of the same type as the "MICHIGAN'S," but with a calorimeter of one-sixth of the grate surface—all other proportions remaining the same, and with the same rate of combustion of Anthracite, the temperature of the furnace averages about 2000° Fahr., and of the products of combustion entering the chimney about 300° Fahr.

As regards the effect of the perforations for the admission of air through the furnace doors, and beneath the bridge walls, I am of opinion that the latter were injurious. They admitted the cold air to the bottom of a large combustion chamber where, as an inspection of the engraving of the boiler will show, there could have been but little temperature, and but little circulation of the gases. The great body of hot gases from the furnace, having much the least specific gravity, passed direct along the upper part of the combustion chamber to the tubes. This current probably did not descend much below the level of the top of the bridge wall. The cold air entering by the perforations considerably below would be re-united in a mass before reaching the heated gases and operate only to cool them.

The perforations in the furnace doors were advantageous, but I am disposed to think on account of the mechanical action of the air admitted by them rather than from its chemical action. In nearly all well proportioned boilers without perforated furnace doors or bridge walls, analyses of their products of combustion show very uniformly the presence of about ten per centum of uncombined oxygen, proving the quantity of air admitted to have been about double that which is chemically required for the combustion. If, therefore, the latter was not complete, it was not owing to any want of air, but to the want of its intimate mixture with the gases in the furnace. If now, we suppose the furnace front to be perforated, then a jet of air will rush in through each perforation with considerable velocity, and by the agitation it produces will cause a more thorough mixture or beating up of the gases. It is known that the more numerous the jets, even to embracing the whole furnace front, the more economical is the result; and the furnace front is the proper place for their introduction, because they then impinge on the gases at their highest temperature, and consequently most favorable condition for chemical union, and because they act upon the whole mass of the gases directly and with the greatest force as they undergo no deflection.

Of the beneficial result attending mere mechanical agitation of the gases in a furnace, every engineer must be aware who has observed the effect of ***back draught*** when using highly bituminous coal. Before the back draught takes place the opaque dull yellow flame of the coal is capped in the furnaces with a dense mass of black smoke, but the instant its concussions commence the smoke vanishes and the opaque yellow flame glows with intense light. The concussions caused by the back draught in boilers are sometimes so violent as to shake a large ship; they act to most thoroughly mix or beat up the gases, and produce as a consequence perfect combustion. The beneficial action of the perforations for admitting air through the furnace doors, I believe due mainly to the same mechanical effect.

If it were practicable to introduce into the furnace, above the bed of incandescent fuel, any mechanical agitator like a revolving wheel with paddles, or its equivalent, I am of opinion that a beneficial effect would follow much greater than can be obtained from any system of admitting air because the mixing of the gases would bé more thorough and the loss of heat due to the superfluous air avoided.

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U. S. SCREW FRIGATES

"MERRIMACK," "WABASH," "MINNESOTA," "ROANOKE,"

AND "COLORADO."

# U. S. SCREW FRIGATES

## "MERRIMACK," "WABASH," "MINNESOTA," "ROANOKE," AND "COLORADO."

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IN the year 1854 Congress passed an act for the construction of six first class steam frigates to be propelled by screws. They were named the "MERRIMACK," "WABASH," "MINNESOTA," "ROANOKE," "COLORADO," and "NIAGARA." Of these all except the latter were designed by the Chief Naval Constructor, JOHN LENTHALL. The "NIAGARA" was designed by MR. GEORGE STEERS, an eminent builder of yachts. The first five vessels were frigate built and armed; their steam power was purely auxiliary and calculated to give them in smooth water, uninfluenced by wind or current, a maximum speed of nine knots per hour. The "NIAGARA" was not a frigate, but an excessively large corvette, carrying guns on her spar-deck only. She was designed with very sharp lines entirely for speed, was furnished with fifty per centum more engine and boiler than the frigates, and her constructor did not restrict himself by any attempt to accommodate her model to the economical use of materials on hand.

The five frigates were built from thoroughly seasoned live oak frames that had been provided for old style sailing frigates. These timbers were the only material available, and in designing the steamers it was more necessary to thoroughly utilize them than to carry out any peculiarity of model, and so completely was this idea realized that the whole of these valuable frames, without waste, was worked into the new construction; the form of the steamers though, consequently, in measure controlled by the shape of frames which had been obtained for another type and dimension of vessel, proved admirably adapted for the purposes intended. The sea-going and fighting qualities of these vessels, and their speed under sail alone was eminently satisfactory. They were the first of their class, and their superiority was so obvious that the British Admiralty immediately imitated them in the construction of the "MERSEY," "ORLANDO," and others of the type.

The five frigates were of nearly the same dimensions and form, as will be observed from the following figures, namely:—

| | "Merrimack." | "Wabash." | "Minnesota." | "Roanoke." | "Colorado." |
|---|---|---|---|---|---|
| Length on mean load-water line from the forward side of rabbet of stem to after side of rabbet of stern-post, in feet and inches, | 256·9 | 262·4 | 264·8½ | 263·8¼ | 263·8¼ |
| Extreme breadth on mean load-water line, in feet and inches, . . | 51·4 | 51·4 | 51·4 | 52·6 | 52·6 |
| Depth from mean load-water line to lower edge of rabbet of keel, in feet, . | 21· | 21· | 21· | 21· | 21· |
| Area of mean load-water line, in square feet, | 10,909·6 | 11,196·2 | 11,818·1 | 11,268·3 | 11,268·3 |
| Area of greatest immersed transverse section, in square feet, . . . | 868·1 | 868·1 | 868·1 | 902·9 | 902·9 |
| Displacement, in tons, . . | 4,635·6 | 4,774·3 | 4,833·4 | 4,772·2 | 4,772·2 |
| Ratio of area of mean load-water line to area of circumscribing parallelogram, . | 0·8274 | 0·8315 | 0·8329 | 0·8139 | 0·8139 |
| Ratio of area of greatest immersed transverse section to area of circumscribing parallelogram, . . . . . | 0·8053 | 0·8053 | 0·8053 | 0·8190 | 0·8190 |
| Ratio of displacement to circumscribing parallelopipedon, . . | 0·5790 | 0·5897 | 0·5917 | 0·5734 | 0·5734 |
| Angle of bow on mean load-water line, | 65 | 65 | 65 | 61 | 61 |
| Angle of stern on mean load-water line, | 77 | 77 | 77 | 73 | 73 |
| Angle of bow on water line at 9½ feet above lower edge of rabbet of keel, | 42 | 42 | 42 | 39 | 39 |
| Angle of stern on water line at 9½ feet above lower edge of rabbet of keel, . | 39 | 39 | 39 | 31 | 31 |
| Angle of dead-rise of greatest immersed transverse section, . . | 14 | 14 | 14 | 12½ | 12½ |

Of these vessels the "MERRIMACK" was the type, the others being only slight modifications of this original.

The "WABASH" is precisely the same as the "MERRIMACK," with the sole exception that at the middle there is inserted a length of 5 feet 7 inches to give more space for machinery and fuel.

The "MINNESOTA" is precisely the same as the "MERRIMACK," with the sole exception that at the middle, in the same manner as in the "WABASH," and for the same reason, there is inserted a length of 7 feet 11½ inches.

The "ROANOKE" and the "COLORADO" are duplicates in all respects, and differ slightly from the others in form. They are fuller in the amidship section, have flatter floors, and sharper ends.

All the frigates have the same spars and sails. The masts are a little varied in position to suit the different lengths of vessel.

They are all furnished with two horizontal, jet condensing engines of the same area and stroke of piston, attached direct to the screw-shaft. The cylinder-valves and valve-gearing are substantially the same in all, consisting of a packed slide, with but trifling lap, worked by a Stephenson Link, and of an independent cut-off slide worked by a separate eccentric, and suppressing the admission of the steam at about one-third the stroke of the piston from the commencement. The "MINNESOTA," "ROANOKE," and "COLORADO," have Penn's trunk engines precisely alike. The engines of the "MERRIMACK" are plain back-

acting with two piston-rods. Those of the "WABASH" are also back-acting, but of the horizontal steeple variety. The boilers are essentially the same in all, in kind, number, dimensions, and arrangement. They are of the vertical water-tube type with the tubes placed above the furnaces according to the patent of D. B. MARTIN.

The propellers of all, with the exception of that of the "MERRIMACK," are alike. They are true screws of 17 feet 4 inches diameter, and 23 feet pitch, with a fraction of pitch of about one-third. The "MERRIMACK" has a Griffiths screw of 17 feet 4 inches diameter, and a mean pitch of 25 feet, with a fraction of pitch of about one-sixth.

The armament and the personnel are the same for all the frigates. The construction, minor details, and general arrangement of their hulls are also alike, and will be found more particularly described under the head of the "MERRIMACK."

In the following pages are given the principal dimensions of the hulls and machinery of use to the engineer; and to them are added tabular abstracts of the performance of each vessel at sea under the actual conditions of ordinary practice. These abstracts embrace *all* the performance recorded in the logs during which the whole data is given; they omit the portion—a very small one—in which the data is incomplete, and they separate the performance under steam alone from that under steam and sails combined.

The number of consecutive hours on each line of the tables is the time during which the wind, sea, and working conditions of the machinery varied but little. The speed was obtained by the common chip log hove hourly by the officer of the deck, and the state of the wind and sea are the mean recorded by him.

The number of double strokes of engine's pistons and of revolutions of the screw made per minute, were taken by a self-registering counter, and entered in the steam-log at the close of each hour by the assistant engineer on watch. He also entered hourly the mean steam pressure in the boilers, the vacuum and throttle opening, the number of pounds of coal expended and its refuse in ashes, &c. Opposite the record of each day's steaming there is pasted in the steam-log an indicator diagram from each end of each cylinder, with the necessary particulars noted on it of revolutions of the screw, steam pressure in boiler, opening of throttle, vacuum in condenser, temperature of hot-well, and state of sea, sail, &c. Nearly all the coal used was Pennsylvania anthracite of good merchantable quality.

A synopsis of the steam-log in the case of each vessel is also given, with a column containing the mean of all the tabular abstracts, and of the results from all the indicator diagrams taken during its cruising. It is believed that the mean of so great a number of observations made under all possible variations of condition, must, from the correction of errors, very closely approximate the truth—too closely to give any error of practical importance. The accuracy of a final result obtained from so extensive a generalization has been frequently proven to the conviction of the writer, and authorizes him to offer it with confidence to others.

There will also be found the maximum performance that can be sustained by each vessel at her mean steaming draught of water, in smooth water and uninfluenced by wind or current, with the machinery in perfect order, and with first quality coal.

It will be observed that this data is very different from that obtained by a run at a measured mile. It is a performance that can be *permanently* sustained under the circumstances, with the loss of fuel incident to "blowing off" the sea water from the boilers to prevent the formation of scale, to normal feeding, and to the cleaning of the fires.

Under the head of "ROANOKE" will be found the data and results of an extended experiment to deter-

mine the evaporative efficiency of the boiler of that vessel with Pennsylvania anthracite. These results will apply with equal truth to the boilers of the other frigates. They are important, as by their means the ultimate condensation in the cylinders can be determined, and as furnishing data for the correct ascertainment of the cost of the steam in fuel, which is the principal element of economy in any system of steam enginery.

# "MERRIMACK."

## HULL.

The frames are of live oak; the distance from centre to centre, or timber and space, is 2 feet 10¼ inches. The lower timbers of the frames are sided 14 inches and the upper timbers 13 inches. The moulding size on the keel is 17 inches—at the floor head 12¾ inches—at the lower port-sill 8½ inches—and at the rail 6 inches. The spaces between the frames are filled in with live oak as high as the turn of the bilge.

There are two sets of diagonal iron braces on the inside of the timbers, and reaching from below the turn of the bilge to the spar-deck clamps. These braces are 4½ inches by ⅞-inch in section, and are fastened to each timber and to each other with 1⅛ inch diameter iron bolts. The stern frame is secured by similar iron braces on the outside of the timbers.

The keel, stem, and stern-post are sided 18½ inches. The garboard strakes are 10 inches in thickness, falling in fair with the bottom plank which is 5 inches in thickness, and thence to the wales which are 7 inches in thickness. The plank above the ports is 4 inches in thickness.

The bilge strakes inside over the butts of the timbers are 6½ inches thick; the berth-deck clamps are 7 inches thick; the gun-deck clamps filling the space between the deck are also 7 inches thick; and the spar-deck clamps are 6 inches in thickness. The ceiling between the thick strakes in the hold is 4½ inches thick. All the planking is of white oak.

The berth-deck beams side 16 inches and mould 13½ inches. The main or gun-deck beams side 17 inches and mould 14½ inches. The spar-deck beams side 16 inches and mould 12½ inches. The berth-deck plank is 4 inches thick; the gun-deck plank is 5 inches thick; and the spar-deck plank is 4½ inches thick. The beams and plank are of Southern yellow pine. The beams have three knees at each end siding from 8½ to 10 inches.

The wales are jogged 1¼ inch over the timbers of the frames, and the clamps are bolted edgewise.

The under side of rail is 3 feet 8 inches, and the top of the hammock rail is 5 feet 8 inches above the spar-deck. The top of hammocks, when stowed, is 7 feet 8 inches above the spar-deck. There are 19 ports on each side of the gun-deck, and the port sills are 20 inches above the deck; the ports are 3 feet 8 inches long and 14 feet 3¼ inches from centre to centre. On this deck there are the galley, and the cabin which is 30 feet long from the stern. On the berth-deck aft there is the ward room, forward of which come the apartments of the warrant officers. Forward of these rooms, at each side, is a coal bunker of moderate size; and the remainder of the deck is appropriated to the crew, which consists of 600 persons. The spar-deck is flush and has no obstructions except the boiler chimney and the hatches. On the after orlop-decks are the cock-pits, two large bread rooms, two sail rooms, two store rooms, and two state rooms for the use of captain's and purser's clerks. The forward orlop has the general store room, two sail rooms, and two bread rooms.

The hull is entirely fastened with copper bolts up to one foot above deep load line, and from that point up with iron bolts. Weight of copper bolts (1½ to ¾-inch diameter) 139,778 pounds; of iron 226,740 pounds.

The stern is round, overhanging, and provided with quarter galleries.

The following are the principal dimensions and proportions of the hull, namely:—

| | |
|---|---|
| Length over all, | 297 feet. |
| Length on mean load-water line from the forward side of rabbet of stem to after side of rabbet of stern-post, | 256 feet 9 inches. |
| Extreme breadth on mean load-water line, | 51 " 4 " |
| Moulded breadth on mean load-water line, | 50 " 2 " |
| Depth from mean load-water line to lower edge of rabbet of keel, | 21 " |
| Depth of keel below lower edge of rabbet, | 18 inches forward and 24 inches aft. |
| Depth from top throat timbers to under side of spar-deck plank amidships, | 33 feet 8 inches. |
| " " " gun-deck plank amidships, | 26 " 2 " |
| " " " berth-deck plank amidships, | 18 " 9 " |
| Displacement to mean load-water line, | 161,930 cubic feet. |
| " " " | 4,635·58 tons. |
| Area of mean load-water line, | 10,909·6 square feet. |
| Displacement per inch of draught at mean load-water line, | 26·026 tons. |
| Area of greatest immersed transverse section to mean load-water line, | 868·1 square feet. |
| Centre of gravity of displacement before middle of the length of mean load-water line, | 2·627 feet. |
| Centre of gravity of displacement below mean load-water line, | 8·300 " |
| Moment of load-water line $\frac{2}{3}\int y^3\,dx =$ | 1,998,130· |
| Height of meta centre above centre of displacement, | 12·462 feet. |
| Angle of bow on mean load-water line, | 65° |
| " stern on mean load-water line, | 77° |
| " bow on water line at 9½ feet above lower edge of rabbet of keel, | 42° |
| " stern on water line at 9½ feet above lower edge of rabbet of keel, | 39° |
| " dead-rise of greatest immersed transverse section, | 14° |
| Ratio of length to breadth on load-water line, | 5·001 to 1·000. |
| " area of greatest immersed transverse section to area of circumscribing parallelogram, | 0·8053 to 1·000. |
| " area of load-water line to area of circumscribing parallelogram, | 0·8274 to 1·000. |
| " displacement to circumscribing parallelopipedon, | 0·5790 to 1·000. |
| Width, in clear, of the screw propeller well, | 7 feet. |
| Height of lower port sills above mean load-water line, amidships, | 9 " |

## TABLE OF SPARS.

| | LENGTHS. | | DIAMETERS. |
|---|---|---|---|
| Mainmast, | 106 feet 5 inches. | At neck. | 3 feet 6 inches. |
| Maintop mast, | 78 " 10 " | Greatest. | 1 " 9 " |
| Maintopgallant mast, | 34 " 0 " | " | 1 " 0¼ " |
| Mainroyal mast, 23 feet, pole 10 feet 10 inches, | 33 " 10 " | | |
| From spar-deck to maintruck, | 222 " 2 " | | |
| Mainyard, | 110 " 4 " | At slings. | 2 " 1¾ " |
| Maintopsail yard, | 83 " 4 " | Greatest. | 1 " 8½ " |
| Maintopgallant yard, | 52 " 3 " | " | 0 " 10¼ " |
| Mainroyal yard, | 35 " 0 " | " | 0 " 7 " |
| Foremast, | 96 " 11 " | " | 3 " 2 " |
| Foretop mast, | 72 " 6 " | " | 1 " 9 " |
| Foretopgallant mast, | 31 " 3 " | " | 1 " 0¼ " |
| Foreroyal mast, 21 feet 3 inches, pole 5 feet, | 26 " 3 " | | |
| From spar-deck to foretruck, | 198 " 9 " | | |
| Foreyard, | 99 " 4 " | | 1 " 11¼ " |
| Foretopsail yard, | 75 " 0 " | | 1 " 6¾ " |
| Foretopgallant yard, | 47 " 0 " | | 0 " 9⅜ " |
| Foreroyal yard, | 31 " 6 " | | 0 " 6½ " |
| Mizzenmast. | 80 " 8 " | | 2 " 8 " |
| Mizzentop mast, | 59 " 3 " | | 1 " 3 " |
| Mizzentopgallant mast, | 25 " 6 " | | 0 " 9 " |
| Mizzenroyal mast, 17 feet 4 inches, pole 4 feet, | 21 " 4 " | | |
| From spar-deck to mizzentruck, | 170 " 6 " | | |
| Cross-jack yard, | 81 " 0 " | | 1 " 5¾ " |
| Mizzentopsail yard, | 61 " 0 " | | 1 " 3 " |
| Mizzentopgallant yard, | 38 " 2 " | | 0 " 7½ " |
| Mizzenroyal yard, | 25 " 6 " | | 0 " 5 " |
| Spanker boom, | 60 " 0 " | | 1 " 1 " |
| Gaff, | 46 " 0 " | | 0 " 8½ " |
| Bowsprit, outboard, | 36 " 0 " | | 3 " 2 " |
| Jibboom, | 27 " 0 " | | 1 " 5 " |
| Flying jibboom, | 23 " 3 " | | 0 " 10½ " |

The centre of the mainmast is 112 feet 4 inches forward of the after side of the stern post. The centre of the foremast is 99 feet from the centre of the mainmast. The centre of the mizzenmast is 75 feet 4 inches from the centre of the mainmast.

## TABLE OF SAILS.

| | SQUARE FEET. | | SQUARE FEET. |
|---|---|---|---|
| Foresail, | 3,827 | Lower studding sails, | each, 2,856 |
| Foretop sail, | 3,525 | Foretop mast studding sails, | " 1,970 |
| Foretopgallant sail, | 1,488 | Maintop mast studding sails, | " 2,190 |

| | SQUARE FEET. | | SQUARE FEET. |
|---|---|---|---|
| Foreroyal, | 693 | Foretopgallant mast studding sails, each, | 924 |
| Mainsail, | 5,100 | Maintopgallant mast studding sails, " | 918 |
| Maintop sail, | 4,396 | Jib, " | 2,408 |
| Maintopgallant sail, | 1,798 | Flying jib, " | 1,780 |
| Mainroyal sail, | 815 | Fore spencer, " | 2,193 |
| Spanker, | 2,139 | Main spencer, " | 2,166 |
| Mizzentop sail, | 2,352 | All storm sails, " | 3,785 |
| Mizzentopgallant sail, | 975 | | |
| Mizzenroyal, | 459 | | |

| | |
|---|---|
| Total surface of all the above sails, | 48,757 square feet. |
| Surface of the ten principal sails, namely:—Jib, Courses, Topsails, Topgallant sails, and Spanker, | 28,008 " |
| Height of centre of effort of the ten principal sails above mean load-water line, | 81·6 feet. |
| Distance of centre of effort of the ten principal sails forward of the centre of displacement, | 8·13 " |
| Ratio of moment of the ten principal sails forward to moment abaft the centre of displacement, | 1·000 to 1·779 |
| Surface of the ten principal sails in proportion to area of load-water line, | 2·574 to 1·000 |
| Surface of the ten principal sails in proportion to area of greatest immersed transverse section, | 32·263 to 1·000 |
| Square feet of the ten principal sails per ton of displacement, | 6·042 to 1·000 |

## ARMAMENT.

SPAR DECK.—Two 10-inch pivot guns weighing without carriages, 12,080 pounds each, one forward and the other aft; also fourteen 8-inch broadside guns weighing without carriages, 7,107 pounds each.

GUN DECK.—Twenty-four 9-inch broadside guns weighing, without carriages, 9,140 pounds each. Total weight of the forty guns, exclusive of carrriages, 343,031 pounds. All the guns are for both solid shot and shells.

Total weight of all objects of ordnance, including the above guns, 774,713 pounds.

## ENGINES.

The engines, two in number, are horizontal and direct-acting, with the connecting rod returned from the crosshead toward the cylinder; each cylinder having two piston-rods, one passing above and the other beneath the crank shaft, and both being secured into lugs forged on the crosshead. The cylinders are placed upon opposite sides of the keelson, in such a manner that the condenser, air-pump, and hot-well of one cylinder are by the side of the other cylinder. Each engine is entirely distinct from the other, and is supported upon a massive cast iron bed-plate extending across the keelson from extreme outboard end of cylinder to outboard end of condenser and hot-well. The cylinder, condenser, and hot-well are cast separately and bolted upon the bed-plate; the main pillow-blocks (two for each engine) are cast upon it; and the air-pump with its channel-ways and valve-chests are cast within it. The air-pump, which is horizontal and double-acting, is lined with brass and fitted with circular gum valves; it is worked direct from

the steam-piston by a piston-rod passing out of the lower part of the inboard cylinder and through a stuffing box. The crosshead works in guides and passes over the top of the air-pump with barely a clearance. The condenser, which is of the common jet kind, is placed on one side of the air-pump but above it, the bottom flange of the condenser being about on a level with the top of the pump: the hot-well is similarly situated on the other side of the air-pump, and the guides are placed between the condenser and hot-well, but farther outboard. The engines are separated by a space 4 feet wide, in which are placed for both engines the feed-pumps, and the eccentrics for working the cylinder-valves. The feed-pumps are of brass, horizontal and single-acting, their axes lie in the same horizontal plane as the axes of the cylinders, and they are worked direct from the end of the crosshead of their respective engines. The steam-valve is a three-ported slide packed on the back which is flat; it lies horizontally on the top of the cylinder and receives its motion from a rock shaft at the outboard end of the cylinder, operated by a Stephenson link: this valve has but very little lap. For the purpose of working the steam expansively, a separate cut-off valve is added, which consists of plain plates moving on a fixed seat of five openings and cutting off the steam by lap. The cut-off valve seat is situated horizontally by the side of the steam-valve seat but 14 inches above it, and the cut-off-valve, like the steam-valve, receives its motion from a rock shaft at the outboard end of the cylinder; this shaft is operated directly by an eccentric. The horizontal distance between the axes of the cylinders in the fore and aft direction of the vessel, is 10 feet 11 inches. The piston packing consists of two thicknesses of cast iron rings set out with springs. The following are the principal dimensions of the engines, namely:—

| | |
|---|---|
| Diameter of the cylinders, | 72 inches. |
| Stroke of the pistons, | 3 feet. |
| Diameter of the piston-rods (two to each cylinder), | 7 inches. |
| Space displacement of both pistons per stroke, exclusive of piston-rods, | 168·040 cubic feet. |
| Clearance, | ¾-inch. |
| Length of steam passage from clearance to steam-valve seat at one end of one cylinder, | 30 inches. |
| Space comprised in clearance and steam passages up to steam-valve seat at one end of both cylinders, | 11 834 cubic feet. |
| Space comprised between steam-valve seat and cut-off-valve seat, exclusive of steam-valve, for both cylinders, | 25·187 " |
| Area of steam-port (5 by 48 inches=), | 240 square inches. |
| Area of exhaust-port (8 by 48 inches =), | 384 " |
| Area of cut-off valve-ports (five ports, each 2⅛ by 24 inches =), | 255 " |
| Diameter of steam-valve stem, | 3¼ inches. |
| Diameter of cut-off valve stem, | 2½ " |
| Diameter of connecting-rod in neck, | 7½ " |
| Length of connecting-rod between centres, | 90 " |
| Diameter of crosshead journal, | 13 " |
| Length of crosshead journal, | 13 " |
| Length of crosshead guide-gib, | 17 " |
| Breadth of crosshead guide-gib, | 9 " |
| Diameter of crank-shaft journals, | Three of 14 and one of 14½ " |
| Length of crank-shaft journals, | Three of 17 and one of 22 " |

| | |
|---|---|
| Diameter of crank-pin journals, . . . . . | 13 inches. |
| Length of crank-pin journals, . . . . . . | 13 " |
| Diameter of screw-shaft journals, . . . . . | 13½ " |
| Length of screw-shaft journals, . . . . . . | 18 " |
| Total width of bearing surface of steam piston, . . . . | 10½ " |
| Total width of packing space of steam piston, . . . . | 7½ " |
| Thickness of the cast iron packing rings (two thicknesses to a piston), | ½-inch. |
| Width of eccentric straps, . . . . . . | 3½ inches. |
| Interior dimensions of condenser above flange on bed-plate, | 74½ by 72 by 17¾ " |
| Diameter of air-pump (double-acting), . . . . . | 22 " |
| Stroke of air-pump piston, . . . . . . | 3 feet. |
| Diameter of air-pump piston-rod, . . . . . . | 3½ inches. |
| Net area of air-pump delivery-valves, . . . . . | 221·7 square inches. |
| Net area of air-pump receiving-valves, . . . . . | 172·2 " |
| Diameter of feed-pump (single-acting), . . . . | 6½ inches. |
| Stroke of feed-pump plunger, . . . . . . | 3 feet. |
| Diameter of injection-valve (one to each condenser), . . | 6½ inches. |
| Diameter of outboard delivery-valve (one for both engines), . . | 18 " |
| Number of collars in thrust bearing, . . . . . | 8. |
| Exterior diameter of collars, . . . . . . | 15½ inches. |
| Interior diameter of collars, . . . . . . | 13½ " |
| Thickness of collars, . . . . . . . | 1½ inch. |
| Space between collars, . . . . . . | 3½ inches. |
| Aggregate thrust surface of collars, . . . . . | 364½ square inches. |
| Net space occupied by the engines in the length of the vessel, | 20 feet 3 inches. |
| Net space occupied by the engines in the breadth of the vessel, . | 22 " 6 " |
| Net space occupied by the engines in height (from bottom of bed-plate), | 12 " |

## BOILERS.

There are four boilers of the vertical water-tube type, with the tubes above the furnaces. The boilers are placed in pairs opposite each other, with a fire room 9 feet wide between them, and extending in the fore and aft direction of the vessel. There is one telescopic chimney in common for all the boilers; it is placed midway the boilers and over the fire room. The two boilers on the same side of the keelson are separated by a space 20 inches wide. Each boiler has a low steam chimney or drum, the four drums surrounding the smoke pipe, whose lower end thus forms one side of them. From the top of these drums the steam is carried to the engines by two steam-pipes—one for each engine. The boilers, with the exception of the tubes, are of iron, and are double riveted everywhere except in the fire surfaces. The bottoms and the ash pits are of $\frac{7}{16}$-inch thick plates; all the other parts are of ⅜-inch thick plates, except the tube plates, which are ½-inch thick.

Each boiler contains four semicircular topped furnaces, and the corners of the ash pits are rounded on a radius of 9 inches. Man-holes, between the spandrels of the furnaces and the lower tube plates, afford easy access to the tops of the furnaces and the bottoms of the tubes; while large hand-holes between the spandrels of the ash pits and the bottoms of the boilers give every requisite facility for cleaning.

The tubes are of brass, and are expanded around the tube plates. The axes of those of the same row are not placed in a straight line lengthwise the boiler, but zig-zag with each other alternately, so as to make the effective width for draught between the rows, crosswise the boiler, much less than the absolute width. The back connexion, tube-box, and tubes for each furnace are entirely separate from those of the other furnaces. Each tube-box contains, lengthwise the boiler, 35 rows of tubes, and crosswise the boiler 10 rows, making 350 tubes in a box, and there are as many boxes as furnaces.

Attached to each boiler and lying beneath the floor plates of the fire room, is a heater composed of a cylindrical cast-iron shell $12\frac{3}{16}$ inches in external diameter, containing 31 brass tubes; each tube is $1\frac{1}{4}$ inch in external diameter, and 13 feet in length. The feed-water is pumped through these tubes into the boiler, and the supersalted water of the boiler is blown out through the space between the tubes and the shell, by means of a surface blow-cock kept wide open and in constant action. The quantity of water blown out is regulated between the heater and the side of the ship, and thus the temperature of the boiler water is maintained constantly upon the surface of the tubes containing the entering feed-water. By this arrangement the feed-water receives, before entering the boiler, an accession of about 30° Fahr. of temperature, without the expenditure of any additional fuel or disadvantage of any kind.

The following are the principal dimensions of the boilers, namely:—

| | |
|---|---|
| Number of boilers, | 4. |
| Length of each boiler at the furnaces (athwartship), | 11 feet 3 inches. |
| " " " top " | 12 " 8 " |
| Breadth of each boiler (fore and aft the vessel), | 14 " |
| Height " " exclusive of steam chimney, | 14 " 3 " |
| " " " inclusive " " | 16 " 5 " |
| Number of furnaces in all the boilers, | 16 |
| Width of each furnace, | 34½ " |
| Length of fire grate, | 7 " 3 " |
| Aggregate area of grate surface in all the boilers, | 333·5 square feet. |
| Height from bottom of ash-pit to crown of furnace, | 3 feet 9 inches. |
| Height from furnace crown to lower tube-plate, | front end 12 inches, back end 9 inches. |
| Width of water-ways between furnaces, | 6 inches. |
| Width of water-bottoms and of water-ways at back connexions, | 7 " |
| Aggregate number of tubes in all the boilers, | 5600 |
| Exterior diameter of tubes, | 2 inches. |
| Interior diameter of tubes, | 1·812 " |
| Total length of each tube, | 39 " |
| Length of each tube in clear of tube-plates, | 38 " |
| Space between the tubes, lengthwise the boilers, | 0·6765" |
| Total length of space occupied by tubes, lengthwise the boiler, | 7 feet 9 inches. |
| Absolute space between the tubes, crosswise the boiler, | 1·3182 inch. |
| Effective space between the tubes for draught, crosswise the boiler, | ¾ " |
| Aggregate absolute space between the tubes, crosswise the boiler, for all the boilers, | 61·222 square feet. |
| Aggregate effective space between the tubes for draught, crosswise the boiler, for all the boilers, | 34·833 " " |

| | | |
|---|---|---|
| Heating surface in all the furnaces, | 976· | square feet. |
| " " the back connexions of all the boilers, | 640· | " " |
| " " " front connexions of all the boilers, | 288· | " " |
| " " tops, sides and bottoms of all the tube boxes, | 1,350 | " " |
| " " all the tubes, measured on their exterior surface, | 9,283 | " " |
| Aggregate area of the heating surface in all the boilers, | 12,537 | " " |
| Diameter of smoke-pipe, | 8 feet. | |
| Height of smoke-pipe above the level of the grates, | 65 " | |
| Steam room in the four boilers, | 2,379 cubic feet. | |
| Ratio of the heating to the grate surface, | 37·592 to 1·000 | |
| " " grate surface to the absolute space between tubes, | 5·447 " " | |
| " " " " " effective space between tubes for draught, | 9·574 " " | |
| " " " " " cross area of the chimney, | 6·635 " " | |
| Weight of water in the four boilers at 60° F. and to 1 foot above tubes, | 180,000 pounds. | |
| Net weight of all the boilers (exclusive of chimney and grate bars), | 257,453 " | |

## SCREW.

The screw is of bronze, and madé, according to the patent of Robert Griffiths, with globular hub and blades adjustible to different pitches. The outline of the blades narrow from the central globe to the periphery of the screw, so that every portion of the blade, radially, has a different fraction of the pitch. When the blades are set at the pitch of 17 feet, they form a *true* screw; but they were never used at that pitch, because it would have required too high a speed of engines and too low an average steam pressure on the pistons. The pitch at which they were orginally set and with which all the steaming recorded in the following "Abstracts of the Steam-Log" was performed, was 26·64 feet at the periphery of the blades, and 23·28 feet at the globe. The mean pitch of the entire blade in function of surface and of the propelling efficiency of the same was just 25 feet.

The calculations of the mean pitch, and of the mean fraction used of the pitch, are made on the postulates that the whole blade being decomposed into an infinite number of *helical lines*, their propelling efficiencies are in the direct ratio of their pitches—the ratio of the square of their distances from the axis—and the ratio of an increase of one-seventh in the propelling efficiency for every doubling of the fraction used of the pitch.

The following dimensions of the screw correspond to the position of the blades when their periphery has a pitch of 26·64 feet:—

| | |
|---|---|
| Diameter of the screw, | 17 feet 4 inches. |
| Diameter of the globular hub, | 5 feet. |
| Mean pitch of the screw in function of surface and of the propelling efficiency of the same, | 25 " |
| Mean fraction used of the pitch in function of surface and of the propelling efficiency of the same, | 0·165 |
| Number of blades, | 2· |
| Length of the screw at the periphery in the direction of the axis, | 1·066 foot. |
| Length of the screw at the longest part (radius 3¼ feet) in the direction of the axis, | 4·162 feet. |

| | |
|---|---|
| Mean angle of blade in function of surface and of the propelling efficiency of the same, | 32° |
| Radius of the centre of pressure of the blade, | 6⅜ feet. |
| Thickness of the blade above fillet, at the radius of 2¼ feet, | 5¾ inches. |
| Thickness of the blade at the periphery, | ¾ inch. |
| Diameter of the pivot attaching blade to globular hub, | 11½ inches. |
| Helicoidal area of the two blades, | 58 square feet. |
| Projected area of the two blades on a plane at right angles to axis, | 46 " " |

## MAXIMUM PERFORMANCE IN SMOOTH WATER, UNINFLUENCED BY WIND OR CURRENT.

The following is the maximum performance that, uninfluenced by wind or current, can be permanently sustained in smooth water with the first quality of steam coal.

The different pressures in the cylinder are the mean of a collation of a large number of indicator diagrams.

| | |
|---|---|
| Vessel's mean draught of water, in feet, | 21·83 |
| Vessel's greatest immersed transverse section, in square feet, | 808·20 |
| Vessel's displacement, in tons, | 4,271·2 |
| Vessel's speed per hour in geographical miles of 6086 feet, | 8·87 |
| Number of double strokes of engine's pistons, and of revolutions of the screw, made per minute, | 46·7 |
| Slip of the screw in per centum of its speed, | 22·94 |
| Portion of the stroke of the piston from the commencement at which the steam is cut off, | 0·307 |
| Steam pressure in the boilers, in pounds per square inch above the atmosphere, | 13·5 |
| Steam pressure in the cylinders at the commencement of the stroke of the piston, in pounds per square inch above zero, | 25·5 |
| Steam pressure in the cylinders at the point of cutting off, in pounds per square inch above zero, | 23·5 |
| Steam pressure in the cylinders at the end of the stroke of the piston, in pounds per square inch above zero, | 9·5 |
| Mean back pressure in the cylinders, in pounds per square inch above zero, | 4·7 |
| Mean vacuum in the condenser in inches of mercury, | 24·5 |
| Mean gross effective pressure on pistons, in pounds per square inch, | 14·2 |
| Mean total pressure on pistons, in pounds per square inch, | 18·9 |
| Gross effective horses power developed by the engines, | 972·525 |
| Total horses power developed by the engines, | 1,294·417 |
| Temperature of feed-water in degrees Fahr., | 137· |
| Pounds of steam discharged per hour from cylinders into condenser, calculated from the pressure of the steam at the end of the stroke of the piston, | 25,239·700 |
| Pounds of steam per hour equivalent to the heat annihilated in the cylinders to produce the total power of the engines, calculated from Joule'e equivalent, | 3,385·030 |

| | |
|---|---|
| Pounds of steam that would have been evaporated per hour, had the heat been so applied that was expended in "blowing off" to maintain the sea-water in the boilers at one and three-fourths time the natural concentration, supposing the boilers to evaporate 10 pounds of water per pound of coal, . . | 5,193·925 |
| Sum of the above three quantities, . . . . . . | 33,818·664 |
| Pounds of water evaporated from temperature of feed-water (137° Fahr.) per hour, supposing 10 pounds of water vaporized by one pound of coal, . . | 42,500· |
| Per centum of the steam evaporated in the boilers not accounted for by the indicator, and by "blowing off," being the per centum which the difference between the quantities on the two preceding lines is of the quantity on the preceding line, | 20·43 |
| Pounds of first quality steam coal consumed per hour, the refuse from it being 12½ per centum, . . . . . . . . . | 4,250· |
| Pounds of coal consumed per hour per square foot of grates, . . . | 12·744 |
| Pounds of coal consumed per hour per gross effective indicated horse power, | 4·371 |
| Pounds of coal consumed per hour per total indicated horse power, . . | 3·283 |

## DISTRIBUTION OF THE POWER DURING THE ABOVE PERFORMANCE.

The pressure required to work the engines and shafting, *per se*, being taken at 1½ pound per square inch of piston, the power thus absorbed is 102·73 horses.

Deducting from the gross effective power of 972·52 horses developed by the engines, this power of 102·73 horses, there remains the power of 869·79 horses applied to the shaft, of which 7½ per centum, or 65·23 horses, is absorbed by the friction of the load.

The power expended in overcoming the cohesive resistance of the water by the screw blades, calculated in the ratio of the square of the velocity, and for a value of 0·45 pound avoirdupois per square foot of helicoidal surface moving in its helical path with a velocity of 10 feet per second, amounts to 38·31 horses.

The powers (65·23 and 38·31 horses) absorbed by the friction of the load and expended in overcoming the cohesive resistance of the water by the screw blades, being deducted from the power (869·79 horses) applied to the shaft, there remains 766·25 horses power expended in the slip of the screw and in the propulsion of the hull. And as the slip of the screw is 22·94 per centum of its speed, the power expended in it is (766·25 × ·2294 =) 175·78 horses, leaving (766·25 — 175·78 =) 590·47 horses expended in the propulsion of the simple hull.

Collecting the foregoing, we have the following distribution of the power, namely:—

| | Horses power. | Per centum. |
|---|---|---|
| Gross effective indicator power developed by the engines, . . . | 972·52 | |
| Power required to work the engines and shafting *per se*, . . . . . | 102·73 | |
| Net power applied to the shaft, . . . . . . . | 869·79 | or 100·00 |
| Power absorbed by the friction of the load, . . . . . . | 65·23 | " 7·50 |
| Power expended in overcoming the cohesive resistance of the water by the screw blades, | 38·31 | " 4·40 |
| Power expended in the slip of the screw, . . . . . . | 175·78 | " 20·21 |
| Power expended in the propulsion of the vessel, . . . . . | 590·47 | " 67·89 |
| Totals, . . . . . . . . . | 9·79 | or 100·00 |

## THRUST OF THE SCREW.

The thrust of the screw during the foregoing maximum performance can be easily calculated from the data in the above distribution of the power. The power required to propel the simple hull is therein found to be 590·47 horses, equal to ($590·47 \times 33000 =$) 19,485,510 pounds raised one foot high per minute. The speed of the vessel was 8·87 geographical miles of 6086 feet per hour or $\left(\frac{8·87 \times 6086}{60} =\right)$ 899·71366 ft. per minute; the resistance of the vessel at this speed, or its equivalent the thrust of the screw, was consequently $\left(\frac{19,485,510}{899·71366} =\right)$ 21,657·457 pounds.

## PERFORMANCE AT SEA UNDER THE CONDITIONS OF ORDINARY PRACTICE.

There will be found in the two following tables abstracts of the entire steam-log of the "MERRIMACK," embracing the whole of her performance recorded therein, of which all the particulars were noted. The performance has been divided into two parts, namely, 1st. That which is contained in the first table, and was done under steam alone. 2d. That which is contained in the second table, and was done under steam and the square sails combined.

Each line of the tables contains the mean performance for the number of consecutive hours stated in the second column. During this time there was but little variation in the weather, state of the sea, rate of combustion of the coal, and manner of using the steam.

The cut-off valve was a fixed slide, and suppressed the admission of the steam into the cylinder at 0·307 of the stroke of the piston from the commencement, that is to say, the steam entered the cylinder during the first 0·307 of the stroke of its piston, and was expanded during the last 0·693 of the stroke.

The throttle-valve was of the usual butterfly kind, turning on a central axis, and was placed in the steam-pipe between the engines. The handle moved on a quadrantal arc concentric with the axis and graduated into equal parts. The numbers in the column headed "Proportion of throttle-valve open," refer to this graduation only, and merely indicate the position of the throttle-valve with relation to this arc, without at all denoting the relative areas of steam-pipe left free for the passage of steam, for these areas are by no means in the proportion of the angular spaces moved over by the throttle-valve in opening.

The number of double strokes made by the piston was registered by a counter worked from the engine. The steam pressure in the boilers was denoted by a mercurial syphon gauge, and the vacuum in the condensers by a closed barometer gauge. The coal and ashes were weighed. A complete set of observations was entered hourly in the steam-log by the engineer officer of the watch, and for each day on which steam was used there was pasted in the log an indicator double diagram from both engines.

The speed of the vessel was taken hourly by the chip-log hove by the officer of the deck. The state of the sea, force and direction of the wind, and course of the vessel, are as recorded by him.

The "slip of the screw" is the difference between the longitudinal speed of its form (product of number of revolutions and pitch) and the speed of the vessel, and is expressed in per centum of the former, except in the few cases when, owing to the effect of the sails, the speed of the latter was the greatest; in these cases the difference is expressed in per centum of the vessel's speed with the minus (—) sign prefixed.

The average draught of water for the entire steaming was 21 feet 10 inches, to which the corresponding

displacement is 4,271·2 tons, with an accompanying greatest immersed transverse section of 808·2 square feet.

The performance recorded in the tables is that which was done during a voyage from the United States to England and back, and from the United States to the north Pacific Ocean and back, and during the vessel's sojourn on the Pacific station.

The vacuum, owing to air leaks, was very poor. The arrangement and proportions of the air-pump were faulty and productive of the same result.

The coal consumed was nearly all Pennsylvania anthracite sent out by the U. S. Navy Department.

The average temperature of the hot-well was 115° Fahr.; this, however, was not the temperature of the feed-water, which, though taken from the hot-well, was pumped through the pipes of the heaters before being delivered into the boilers; and as these pipes were surrounded by the water *continuously* blown from the boilers through a small surface blow-cock to prevent the formation of scale, the feed-water received from this blown out water sufficient heat to raise its temperature to an average of 137° Fahr., at which it entered the boilers. The heaters were situated by the side of the keelson, in front of the boilers, and beneath the fire-room floor.

The density of the boiler water was maintained at one and three-quarter time the density of sea water. The heating surfaces of the boilers being everywhere easily accessible to cleaning tools, were kept perfectly free from scale and at their maximum evaporative efficiency.

The two tables above referred to contain the performance of the vessel, in detail, under the conditions of wind, sea, and sail stated; the means of the quantities in these tables will be found in the table which follows them, under the heading of "Synopsis of the Steam-log of the U. S. Screw Frigate MERRIMACK," together with the calculated results therefrom. In making these calculations, the tables of REGNAULT are used for the sensible and latent heat of steam. The density of the steam is ascertained by FAIRBAIRN'S formulæ, the weight of a cubic foot of water at 62° Fahr. being taken at 62·321 pounds. The indicator diagrams analyzed, were several hundred in number, taken under every variety of weather, draught of water, &c.

## ABSTRACT OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "MERRIMACK," EMBRACING ALL HER PERFORMANCE UNDER STEAM ALONE.

| DATE. | Number of consecutive hours. | Course of the Vessel. | WIND. Direction. | WIND. Kind. | STATE OF THE SEA. | Speed of the Vessel per hour in geographical miles of 6086 feet. | Number of double strokes of engines' pistons, and of revolutions of the screw, made per minute. | Steam pressure in Boilers, in pounds per sq. inch above the atmosphere. | Vacuum in Condensers, in inches of Mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Anthracite consumed per hour. | Per centum of the Coal lost, in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1856. | | | | | | | | | | | | | |
| Feb. 28 & 29, | 48 | W. | N. W. by N. | Mod. breeze. | Mod'ate. | 6·083 | 38·45 | 11·5 | 21·2 | 0·25 | 35·81 | 2979 | 21⅓ |
| March 1 & 2, | 32 | S. W. | N. N. W. | Light " | Smooth. | 6·125 | 36·57 | 12·4 | 20·3 | 0·20 | 32·08 | 3056 | " |
| May 9, | 8 | E. S. E. | S. E. | " " | " | 6·875 | 41·25 | 9 0 | 22·0 | 0·62 | 32·38 | 3370 | 17 |
| " 16, 17, 18, 19, 20, 21 | 84 | S. | S. | Gentle " | " | 5·738 | 34·92 | 10·2 | 22·0 | 0·18 | 33·33 | 3161 | " |
| August 14 & 15, | 34 | . . | Ahead. | Light airs. | " | 7·000 | 40·69 | 10·5 | 23·0 | 0·25 | 30·20 | 3146 | 19 |
| Sept. 16, | 24 | E. | S. E. | " " | " | 6·833 | 35·88 | 13·1 | 22·0 | 0·12 | 22·78 | 3474 | 18 |
| Nov. 1, | 9 | . . | Forward beam. | " breeze. | Rough. | 5·111 | 34·38 | 16·8 | 21·0 | 0·20 | 39·60 | 2900 | 19¾ |
| Dec. 4 & 5, | 32 | S. by E. | S. S. W. | Mod. " | Mod'ate. | 6·500 | 39·75 | 15·4 | 21·7 | 0·20 | 33·65 | 4147 | 19 |
| 1857. | | | | | | | | | | | | | |
| March 10, | 21 | N. W. by N. | N W. | Light " | Smooth. | 7·000 | 44·09 | 14·7 | 20·5 | 0·16 | 35·59 | 4254 | 20½ |
| " 12 & 13, | 40 | N. N. W. | N. N. E. | Strong " | Rough. | 2·600 | 39·60 | 18·2 | 21·0 | 0·16 | 78·35 | 4500 | " |
| " 14, | 12 | E. by N. | N. E. by N. | " " | Heavy. | 2·167 | 30·35 | 17·2 | 17·5 | 0·12 | 71·04 | 3770 | " |
| " 27, | 8 | N. E. | S. E. by E. | Light airs. | Smooth. | 7·875 | 42·50 | 19·4 | 20·0 | 0·14 | 24·82 | 4507 | 13 |
| " 30, | 12 | N. W. | N. by W. | Gentle breeze. | " | 7·333 | 42·08 | 19·2 | 20·0 | 0·20 | 29·29 | 4065 | " |
| October 19 & 20, | 32 | S. E. | S. E. by E. | " " | " | 5·875 | 34·19 | 16·2 | 19·8 | 0·33 | 36·21 | 2784 | 15 |
| Nov. 11 & 12, | 30 | S. by E. | S. E. by E. | Light " | " | 6·267 | 36·76 | 15·0 | 20·0 | 0·25 | 30·83 | 3367 | " |
| " 14, 15, 16, & 17, | 75 | S. | S. S. E. | " " | " | 4·840 | 38·98 | 15·6 | 21·0 | 0·25 | 42·21 | 3228 | " |
| 1858. | | | | | | | | | | | | | |
| Jan. 8 & 9, | 36 | S. by E. | S. W. | " " | " | 5·917 | 38·05 | 17·5 | 21·0 | 0·82 | 36·90 | 3681 | 14¾ |
| " 11 & 12, | 38 | W. S. W. | W. by N. | Gentle " | Mod'ate. | 4·878 | 32·89 | 16·9 | 19·3 | 0·60 | 39·82 | 3395 | " |
| " 13 & 14, | 36 | W. S. W. | W. S. W. | " " | " | 5·083 | 36·89 | 18·1 | 22·8 | 0·27 | 44·09 | 3829 | " |
| " 15 & 16, | 32 | N. W. | S. W. | Light " | Smooth. | 6·812 | 40·11 | 20·0 | 22·1 | 0·25 | 31·09 | 3976 | " |
| 1859. | | | | | | | | | | | | | |
| May 7, 8, 9, 10, 11, 12, 13, 14, 15, | 188 | S. | S. | Gentle " | Gentle. | 4·192 | 34·98 | 12·0 | 18·7 | 1·00 | 51·81 | 3330 | 21¼ |
| " 17, 18, & 19, | 50 | S. E | S. E. | " " | " | 4·050 | 34·85 | 12·2 | 20·1 | 1·00 | 52·84 | 3359 | 21½ |
| " 23 & 24, | 29 | S. E | S. E. | Light " | " | 4·207 | 34·63 | 11·1 | 20·0 | 0·33 | 50·71 | 4202 | " |
| Nov. 14 & 15, | 25 | S. W. | S. by W. | " " | Mod'ate. | 5·000 | 39·09 | 12·0 | 20·4 | 1·00 | 48·10 | 3000 | 24½ |
| " 29 & 30, | 36 | N. N. E. | N. N. W. | " " | Gentle. | 4·806 | 36·26 | 10·0 | 20·5 | 0·83 | 46·22 | 3358 | " |
| Dec. 12 & 13, | 31 | N. by E. | N. by E. | " " | Smooth. | 5·549 | 36·80 | 10·0 | 20·5 | 0·41 | 38·82 | 2603 | " |
| " 24, | 7 | E. S. E. | E. by N. | Mod. " | Mod'ate. | 5·571 | 38·66 | 10·0 | 20·0 | 1·00 | 41·53 | 2650 | 12⅔ |
| 1860. | | | | | | | | | | | | | |
| Jan. 1, 2, & 3, | 33 | N. by E. | N. E. | Light " | Smooth. | 6·363 | 36·17 | 10·0 | 19·0 | 0·75 | 28·63 | 3150 | " |
| " 13 & 14, | 28 | N. | N. E. | " airs. | " | 5·875 | 34·68 | 10·0 | 19·2 | 0·62 | 37·11 | 3193 | " |
| " 27, | 8 | N. W. by N. | W. | " " | " | 6·125 | 42·37 | 10·2 | 20·0 | 0·88 | 41·36 | 2800 | " |
| " 31, | 24 | N. N. W. | S. S. E. | " breeze. | " | 6·542 | 37·00 | 10·0 | 19·0 | 0·35 | 28·26 | 3068 | " |
| Feb. 3 & 4, | 39 | N. W. | N. by W. | Gentle " | Rough. | 4·462 | 36·34 | 12·1 | 19·1 | 0·90 | 50·18 | 3572 | " |
| Means, | | | 29° from Ahead. | Gentle breeze. | Gentle. | 5·249 | 36·54 | 12·8 | 20·4 | 0·50 | 41·74 | 3401 | 18⅓ |

## ABSTRACT OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "MERRIMACK," EMBRACING ALL HER PERFORMANCE UNDER STEAM AND SAILS COMBINED.

| DATE. | Number of consecutive hours. | Course of the Vessel. | WIND. | | STATE OF THE SEA. | Speed of the Vessel per hour, in geographical miles of 6086 feet. | Number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute. | Steam pressure in Boilers, in pounds per sq. inch above the atmosphere. | Vacuum in Condensers, in inches of mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Anthracite consumed per hour. | Per centum of the Coal lost, in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Direction. | Kind. | | | | | | | | | |
| 1856. | | | | | | | | | | | | | |
| Feb. 25 & 26, | 32 | . . | On Quarter. | Strong breeze. | Rough. | 10·656 | 38·50 | 14·7 | 21·5 | 0·47 | −10·18 | 2838 | 21⅛ |
| " 27, | 24 | S. W. | N. W. | " " | " | 5·208 | 29·00 | 11·1 | 21·0 | 0·14 | 27·13 | 2544 | " |
| May 10, | 8 | S. by E. | W. | Mod. " | Smooth. | 8·125 | 39·00 | 5·8 | 22·0 | 0·38 | 15·47 | 3155 | 17 |
| " 16, | 12 | S. S. E. | S. S. W. | " " | " | 7·167 | 37·90 | 7·7 | 22·0 | 0·28 | 23·27 | 3330 | " |
| August 16, | 12 | W. | N. W. | Gentle " | " | 6·750 | 39·00 | 8·8 | 22·0 | 0·25 | 29·78 | 2927 | 19 |
| Sept. 9, 10, & 11, | 57 | E. | S. | " " | " | 9·000 | 40.82 | 13·5 | 23·4 | 0 13 | 10·54 | 3490 | 18 |
| " 13, | 12 | E. by S. | S. | " " | " | 8·500 | 39·00 | 13·1 | 22·5 | 0·18 | 1[illegible]·57 | 2805 | " |
| " 17, | 12 | E. | S. S. E. | Mod. " | " | 8·250 | 38·74 | 13·7 | 21·0 | 0·13 | 13·59 | 3529 | " |
| " 24, | 10 | E. | On Quarter. | Gentle " | " | 9·000 | 41·78 | 15·2 | 22·0 | 0·16 | 12·60 | 3900 | " |
| Oct. 30, | 10 | W. S. W. | S. by W. | Strong " | Rough. | 7·000 | 44.30 | 15·7 | 22·0 | 0·25 | 35·89 | 4028 | 19¾ |
| " 31, | 24 | W. S. W. | N. W. | Gentle " | Moderate. | 6·417 | 36·94 | 15·0 | 23·0 | 0·20 | 29·51 | 3378 | " |
| Nov. 13, 14, 15, & 16, | 68 | S. W. by S. | N. W. | Mod. " | Smooth. | 9·177 | 44·66 | 17·2 | 21·5 | 0·20 | 16·62 | 4000 | " |
| Dec. 29 & 30, | 16 | W. by S. | On Quarter. | Fresh " | Moderate. | 10·125 | 40·50 | 18·7 | 19·6 | 0·18 | −1·41 | 3500 | 27 |
| 1857. | | | | | | | | | | | | | |
| March 11 & 12, | 32 | N. W. by N. | S. by W. | Mod. " | Smooth. | 8·875 | 41·39 | 15·5 | 20·2 | 0·15 | 18·00 | 4139 | 20½ |
| " 13, | 12 | N. by W. | N. E. | Fresh " | Rough. | 8·417 | 42·08 | 16·7 | 18·0 | 0·20 | 18·84 | 4488 | " |
| " 28 & 29, | 48 | N. E. | N. N. W. | Mod. " | Smooth. | 8·833 | 42·74 | 18·0 | 20·0 | 0·16 | 16·15 | 3898 | 13 |
| Oct. 18, | 24 | S. E. | W. | Light " | " | 7·875 | 38·58 | 16·5 | 20·0 | 0·17 | 17·18 | 3000 | 15 |
| Nov. 13 & 14, | 20 | S. ½ E. | E. | Gentle " | " | 8·000 | 39·39 | 16·6 | 21·1 | 0·25 | 17·59 | 2987 | " |
| " 19 & 20, | 23 | S. by W. | S. E. | " " | " | 6·130 | 38·76 | 17·6 | 22·6 | 0·42 | 35·81 | 3562 | " |
| 1858. | | | | | | | | | | | | | |
| Jan. 9 & 10, | 36 | S. by W. | N. W. by W. | Fresh " | Rough. | 7·588 | 39·64 | 16·9 | 20·5 | 0·37 | 22·38 | 3875 | 14¾ |
| " 12 & 13, | 20 | W. N. W. | S. W. | Strong " | Moderate. | 6·050 | 37·21 | 15·0 | 23·3 | 0·38 | 34·03 | 3581 | " |
| " 17, | 7 | N. W. by N. | S. W. by W. | " " | " | 8·857 | 40·34 | 17·4 | 22·5 | 0·20 | 10·91 | 2674 | " |
| " 18, | 24 | N. N. W. | W. by N. | Gentle " | " | 6·750 | 40·11 | 20·3 | 23·0 | 0·37 | 31·72 | 4025 | " |
| 1859. | | | | | | | | | | | | | |
| Nov. 26, 27, 28, & 29, | 63 | E. | N. N. W. | Fresh " | Rough. | 6·762 | 38·75 | 11·4 | 20·0 | 0·46 | 29·19 | 2906 | 24½ |
| Dec. 1, 2, & 3, | 43 | N. N. E. | W. N. W. | Mod. " | Moderate. | 7·872 | 40·12 | 10·0 | 20·9 | 0·34 | 25·44 | 2966 | " |
| " 14, 15, & 16, | 40 | N. | E. | Light " | Smooth. | 6·175 | 35·88 | 10·0 | 19·7 | 0·20 | 30·17 | 2661 | " |
| " 25, | 24 | E. by S. | N. E. by N. | Mod. " | Rough. | 5·459 | 36·06 | 10·0 | 19·7 | 0·62 | 38·58 | 3415 | 12⅝ |
| " 26, | 8 | E. | N. E. by N. | Fresh " | " | 6·500 | 34·54 | 10·0 | 19·8 | 0·20 | 23·65 | 3150 | " |
| 1860. | | | | | | | | | | | | | |
| Jan. 15, | 16 | N. by W. | E. | Gentle " | Smooth. | 6·625 | 36·04 | 10·0 | 18·8 | 0·25 | 25·42 | 3238 | " |
| Feb. 1, | 10 | N. N. W. | S. W. by W. | Fresh " | Rough. | 7·800 | 38·30 | 11·0 | 17·8 | 0·25 | 17·37 | 3164 | " |
| " 2, | 17 | N. N. W. | N. E. | Gentle " | " | 4·353 | 35·14 | 10·6 | 19·8 | 0·75 | 49·74 | 3360 | " |
| Means, | | | 90° from Ahead. | Mod. breeze. | Moderate. | 7·670 | 39·34 | 12·5 | 21·0 | 0·28 | 20·90 | 3392 | 18½ |

## SYNOPSIS OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "MERRIMACK."

| | Under Steam Alone. | Under Steam and Square Sails combined. | Mean of the two preceding Columns. |
|---|---|---|---|
| Total number of hours, | 1136· | 764· | 1,900· |
| Kind of wind, | Gentle breeze. | Moderate breeze. | . . |
| Angle made from ahead by the wind with the line of the vessel's keel, | 29° | 90° | 58°·5 |
| State of the sea, | Gentle. | Moderate. | . . |
| Speed of the Vessel per hour, in geographical miles of 6086 feet, | 5·249 | 7·670 | 6·222 |
| Number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute, | 36·54 | 39·34 | 37·67 |
| Slip of the Screw, in per centum of its axial speed, | 41·74 | 20·90 | 32·99 |
| Steam pressure in the boilers, in pounds per square inch above the atmosphere, | 12·8 | 12·5 | 12·7 |
| Vacuum in Condensers, in inches of mercury, | 20·4 | 21·0 | 20·6 |
| Proportion of Throttle-valve open, | 0·50 | 0·28 | 0·415 |
| Temperature in degrees Fahr. of the Hot-well, | 110° | 110° | 110° |
| Temperature in degrees Fahr. of the Feed-water (after passing through the Heaters), | 137° | 137° | 137° |
| Fraction of the Stroke of the Piston completed when the steam was cut off, per Indicator, | 0·807 | 0·307 | 0·307 |
| **Steam Pressure in Cylinders, per indicator.** | | | |
| In pounds per square inch above zero at commmencement of Stroke of Piston, | | | 23·5 |
| In pounds per square inch above zero at point of cutting off the Steam, | | | 21·5 |
| In pounds per square inch above zero at end of Stroke of Piston, | | | 9·0 |
| In pounds per square inch above zero against the Piston during its stroke, | | | 6·9 |
| Mean Gross Effective Pressure in pounds per square inch on Piston during its stroke, | | | 10·1 |
| Mean Total Pressure in pounds per square inch on Piston during its stroke, | | | 17·0 |
| Gross Effective Horses Power developed by the Engines, | | | 557·972 |
| Total Horse Power developed by the Engines, | | | 989·161 |
| Pounds of Anthracite consumed per hour, | 3,401· | 3,392· | 3,398· |
| Pounds of Refuse from Anthracite per hour, in ashes, clinkers, and dust, | 619· | 628· | 623· |
| Per centum of Refuse from the Anthracite, | 18·2 | 18·5 | 18·3 |
| Pounds of Combustible consumed per hour, | 2,782· | 2,764· | 2,775· |
| Pounds of Anthracite consumed per hour per square foot of grate, | 10·198 | 10·171 | 10·189 |
| " Combustible " " " | 8·342 | 8·288 | 8·321 |
| Pounds of Anthracite consumed per hour per Gross Effective Horse Power, | | | 6·090 |
| Pounds of Anthracite consumed per hour per Total Horse Power, | | | 3·618 |
| Pounds of Combustible consumed per hour per Gross Effective H. P., | | | 4·973 |
| Pounds of Anthracite consumed per hour per Total Horse Power, | | | 2·955 |
| **Evaporated from temperature of Feed-water.** | | | |
| Pounds of Steam discharged per hour from Cylinders into Condenser, calculated from the pressure of the steam at the end of the Stroke of the Piston, | . . | . . | 19,336·245 |
| Pounds of Steam per hour equivalent to the Heat annihilated in the Cylinders to produce the Total Power of the Engines calculated from JOULE's equivalent, | . . | . . | 2,451·632 |
| Pounds of Steam that would have been evaporated per hour, had the Heat been so applied that was expended in "blowing off," to maintain the Sea-water in the Boilers at one and three-fourths time the natural concentration, supposing the Boilers to evaporate 9½ pounds of water per pound of Anthracite, | . . | . . | 3,896·317 |
| Sum of the above three quantities, | . . | . . | 25,684·194 |
| Pounds of Water evaporated from temperature of Feed-water (137° Fahr.) per hour, supposing 9½ pounds of water vaporized by one pound of Anthracite, | . . | . . | 32,281·000 |
| Per centum of the Steam evaporated in the Boilers not accounted for by the Indicator, and by "blowing off," being the per centum which the difference between the quantities on the two preceding lines is of the quantity on the preceding line, | . . | . . | 20·44 |

# "WABASH."

## HULL.

In all other respects, except the following dimensions and proportions, the hull of the "WABASH" is the same as that of the "MERRIMACK."

| | |
|---|---|
| Length over all, from after side of taffrail to end of billet, | 301 feet. |
| Length on mean load-water line from the forward side of rabbet of stem to after side of rabbet of stern-post, | 262 feet, 4 inches. |
| Extreme breadth on mean load-water line, | 51 " 4 " |
| Moulded breadth on mean load-water line, | 50 " 2 " |
| Depth from mean load-water line to lower edge of rabbet of keel, | 21 " |
| Depth of keel below lower edge of rabbet, | 2 " |
| Depth from top throat timbers to underside of spar-deck plank amidships, | 33 " 8 " |
| Depth from top throat timbers to underside of gun-deck plank amidships, | 26 " 2 " |
| Depth from top throat timbers to underside of berth-deck plank amidships, | 18 " 9 " |
| Height of lower port-sill above rabbet of keel amidships, | 30 " |
| Height of upper port-sill above rabbet of keel amidships, | 37 " 4 " |
| Displacement to mean load-water line, | 166,777 cubic feet. |
| Displacement to mean load-water line, | 4,774·33 tons. |
| Area of mean load-water line, | 11,196·2 square feet. |
| Displacement per inch of draught at mean load-water line, | 26·71 tons. |
| Area of greatest immersed transverse section to mean load-water line, | 868·1 square feet. |
| Angle of bow on mean load-water line, | 65° |
| Angle of stern on mean load-water line, | 77° |
| Angle of bow on water line at 9½ feet above lower edge of rabbet of keel, | 42° |
| Angle of stern on water line at 9½ feet above lower edge of rabbet of keel, | 39° |
| Angle of dead-rise at greatest immersed transverse section, | 14° |
| Ratio of length to breadth on load-water line, | 5·110 to 1·000 |
| Ratio of area of greatest immersed transverse section to area of circumscribing parallelogram, | 0·8053 |
| Ratio of area of load-water line to area of circumscribing parallelogram, | 0·8315 |
| Ratio of displacement to circumscribing parallelopipedon, | 0·5897 |

The space occupied by the machinery and coal, is a length of 60 feet by the entire height and breadth of the vessel below the berth-deck.

The length from the forward side of the engines to the after side of the stern-post is 135 feet.

## ENGINES.

There are two horizontal, condensing, steeple engines (with a horizontal wrought iron steeple or harp from the small extremity of which two connecting rods are returned to the crank-pin). The two cylinders are placed on the same side of the keelson, and have their condensers, hot-wells, and pumps opposite. There is a free passage way between and around the engines.

The cylinder, condenser, and hot well of each engine, are placed on a bed-plate 7 feet 6 inches wide, 24 inches deep at the sides and 13 inches deep at the centre, extending 22 feet across the vessel measured from the outer end of the cylinder to the outer end of the condenser.

The air-pump is placed inside the bed-plate; it is horizontal and double-acting, and is worked by a rod from the bottom of the yoke or broad part of the steeple. Each engine has a feed and a bilge-pump which are horizontal, single-acting, and worked by rods from the side of the steeple: these pumps lie above the air-pump.

There is one piston-rod to each cylinder, and it is secured to the centre of the yoke of the steeple. The bottom of the yoke is supported by and slides upon a smooth plate. The crosshead is secured to the smaller end of the steeple and works between guides: from this crosshead there are returned on each side the steeple a connecting rod, making two connecting rods to each engine; the crank-shaft operated by these rods revolves within the steeple with the cranks on each side. The crosshead guides are above the level of all the pumps and lie between the condenser and hot-well which are chambers each 93 inches high above the bed-plate (66 inches above axis of crank-shaft) and 23 by 43 inches in horizontal projection.

The cylinder-valves are placed horizontally on the upper side of the cylinder. The steam valve is a packed flat slide; the cut-off valve works on its back and consists of two plates adjustable by right and left hand screws. The steam-valve is worked by a Stephenson Link from a rock shaft; the cut-off valve is worked by an independent eccentric from a rock shaft also. The eccentrics for the steam-valves are between the engines, and for the cut-off valves on the outside of the engines.

In addition to the thrust collars, the shaft is provided with the Parry roller thrust.

The following are the principal dimensions of the engines; namely:—

| | |
|---|---|
| Diameter of cylinders, . . . . . . | 72 inches. |
| Diameter of piston-rod (one to each cylinder), . . . | 7¼ " |
| Stroke of pistons, . . . . . . . | 3 feet. |
| Space displacement of both pistons per stroke, exclusive of rod, . . | 168·840 cubic feet. |
| Space comprised between the piston at the end of its stroke and the steam-valve, at one end of both cylinders, . . . . . | .9·356 " |
| Space comprised between the faces of steam and cut-off valves, at one end of both cylinders, . . . . . . . | 1·384 " |
| Clearance, . . . . . . . . | ¾-inch. |
| Area of the steam-port (5¼ by 48 inches), . . . . . | 252 square inches. |
| Area of the exhaust-port (7 by 48 inches), . . . . | 336 " |
| Area of the passage through the steam-valve (4 by 46 inches), . . | 184 " |
| Length of the passage through the steam-valve, . . . . | 6½ inches. |

| | |
|---|---|
| Lap of the steam-valve, | 1 inch. |
| Travel of the steam-valve, | 12½ inches. |
| Diameter of each connecting-rod at neck (two to each engine), | 5½ inches at crosshead and 6 inches at crank pin. |
| Length of connecting-rod, | 7 feet. |
| Diameter of crosshead journals (two to each engine), | 6 inches. |
| Length of crosshead journals (each), | 7 " |
| Area of crosshead guide gibs (7 by 18 inches), | 126 square inches. |
| Diameter of air-pump, | 22 inches. |
| Stroke of air-pump piston, | 3 feet. |
| Space displacement of air-pump piston per stroke, | 7·92 cubic feet. |
| Net area of air-pump receiving-valves, | 208 square inches. |
| Net area of air-pump delivery-valves, | 208 " |
| Diameter of feed-pump (also of bilge-pump), | 6½ inches. |
| Stroke of feed-pump piston (also of piston of bilge-pump), | 3 feet. |
| Space displacement of feed-pump piston per stroke, | 0·69132 cubic feet. |
| Diameter of crank-shaft journals, | 14 inches. |
| Length of crank-shaft journals, | 18 " |
| Diameter of crank-pin journals (two to each engine), | 15 " |
| Length of crank-pin journals, | 8 " |
| Diameter of screw-shaft journals, | 13½ " |
| Length of screw-shaft journals, | 18 " |
| Length of packing space of steam piston, | 7 " |
| Length of total bearing of steam piston, | 10 " |
| Length of packing space of air-pump piston, | 6 " |
| Length of total bearing of air-pump piston, | 8 " |
| Width of eccentric strap, | 4 " |
| Number of collars in thrust bearing, | 9. |
| Outside diameter of collars, | 17 " |
| Inside diameter of collars, | 14 " |
| Thickness of the collars, | 2 " |
| Distance between axes of cylinders, | 10½ feet. |
| Extreme length of the engines (fore and aft the ship), | 19 " |
| Extreme breadth of the engines (athwartship), | 22 " |
| Extreme height of the engines (from bottom of bed-plate), | 10 " |

## FINISHED WEIGHT OF ENGINES.

| | |
|---|---|
| Wrought iron, | 111,267· pounds. |
| Cast iron, | 212,917· " |
| Brass, including 4,886 pounds in feed and blow-pipes, | 42,189· " |
| Steel, | 814· " |
| Copper forgings, | 1,432· " |
| Copper pipes, | 13,137· " |

| | |
|---|---|
| Plate iron in coal bunkers, bulkheads, and shaft alley, | 50,877· pounds. |
| Ventilators, stairs, floors, deck-plates, &c., principally cast iron, | 37,485· " |
| Total, | 470,118 pounds. |

## SCREW.

One true screw of brass. The back and front edges of the blades taper in a straight line from hub to periphery. It hoists up through a well in the stern.

| | |
|---|---|
| Diameter of the screw, | 17 feet 4 inches. |
| Diameter of the hub at front and back, | 21 and 18 " |
| Pitch, | 23 feet. |
| Length of each blade in direction of axis, on hub, (tapering in a straight line to a,) | 42 inches. |
| Length of each blade in direction of axis, at periphery, | 36 " |
| Thickness of blade at centre, above fillet of hub, | 10 " |
| Thickness of blade at periphery, | 1 inch. |
| Radius with which the corners of the blades are rounded off, | 18 inches. |
| Number of blades, | 2. |
| Helicoidal area of the two blades, | 87· square feet. |
| Projected area of the two blades on a plane at right angles to axis, | 67· " |
| Mean fraction of pitch used, in function of surface and propelling efficiency of the same, | 0 273 |
| Diameter of forward journal of hub, | 20 inches. |
| Length of forward journal of hub, | 12 " |
| Diameter of after journal of hub, | 15 " |
| Length of after journal of hub, | 12 " |

## FINISHED WEIGHTS OF SCREW AND ITS HOISTING APPARATUS.

| | |
|---|---|
| Screw, of brass, | 20,020 pounds. |
| Hoisting apparatus, of brass, | 9,535 " |
| Guides for hoisting apparatus, of brass, | 4,905 " |
| Total, | 34,460 pounds. |

## BOILERS.

There are four boilers of the vertical water-tube kind with the tubes placed above the furnaces. The boilers are in pairs, situated opposite each other with a fire room 9 feet wide extending between them in the fore and aft direction of the vessel. There is one chimney for the four boilers; it is telescopic and placed at the centre of and above the fire room; its lower portion is surrounded by a steam chimney or drum. The tops of the furnaces and the bottoms of the ash-pits are semicircular.

The tubes of each row in the direction of the length of the furnace are not arranged in the same straight line, but are placed staggered or alternately zigzagged. The two straight lines in which the centres of the same row of tubes are alternately placed, are parallel and $\frac{1}{4}$-inch apart.

The boilers are double riveted in all seams not exposed to the action of the fire. The tube-plates are $\frac{1}{2}$-inch thick; the bottoms of the boilers and of the ash-pits are of $\frac{7}{16}$ths inch thick plate; all other portions

are of ⅜ths inch thick plate. The tubes are of brass, and are expanded on one side of their plates and riveted over on the other.

Between the two boilers on the same side of the keelson, there is a passage-way of 20 inches width.

The boilers are provided with heaters of the same dimensions and arrangement described under the head of boilers of the "MERRIMACK." The furnace doors were not pierced with holes for air admission.

The following are the principal dimensions of the boilers; namely:—

| | |
|---|---|
| Number of boilers, | 4. |
| Length of each boiler, athwartships, at furnaces, | 11 feet. |
| Length of each boiler, athwartships, at top, | 12 " |
| Breadth of each boiler, fore and aft the vessel, | 15 " |
| Height of each boiler, exclusive of steam chimney, | 13½ " |
| Height of each boiler, inclusive of steam chimney, | 15¾ " |
| Number of furnaces in each boiler, | 5. |
| Breadth of each furnace, | 29 inches. |
| Length of grate bars, | 7 feet. |
| Height from bottom of ash-pit to crown of furnace, | 45 inches. |
| Width of water-bottoms, and of legs between furnaces, | 6 " |
| Width of water-legs between wing furnaces and shell of boiler, | 5½ " |
| Width of water-legs at front and back of boiler, | 5 " |
| Distance between the crown of the furnace and the lower tube-plate at front, | 12 " |
| Distance between the crown of the furnace and the lower tube-plate at back, | 10 " |
| Length of space occupied by tubes (in direction of length of furnace), | 94 " |
| Number of rows of tubes lengthwise the furnace, | 34. |
| Number of rows of tubes crosswise the furnace, | 8. |
| Total number of tubes in one boiler, | 1360. |
| Distance in the clear between the tubes lengthwise the furnace, | 0·788 inch. |
| Distance in the clear between the tubes crosswise the furnace, | 1⅜ " |
| Distance for direct draught between the tubes crosswise the furnace, | 1·1944 " |
| Calorimeter for the four boilers, absolute, | 63·1944 square feet. |
| Calorimeter for the four boilers, for direct draught, | 53·4722 " |
| Length of tubes, extreme, | 36 inches. |
| Length of tubes between plates, | 35 " |
| Exterior diameter of tubes, | 2 " |
| Interior diameter of tubes, | 1·73 " |
| Diameter of smoke-pipe, | 8 feet. |
| Height of smoke-pipe above the level of the grates, | 61 " |
| Total area of grate surface in the four boilers, | 338⅓ square feet. |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Area of heating surface of the furnaces of the four boilers, | | | | | 810·00 square feet. | | |
| " | " | " | " | back connexions | " | " 740·00 | " " |
| " | " | " | " | front connexions, | " | " 232·00 | " " |
| " | " | " | " | tube boxes, | " | " 1,526·00 | " " |
| " | " | " | " | tubes, | " | " 8,544·88 | " " |

Total area of heating surface in the four boilers, . . 11,852·88 square feet.

| | |
|---|---|
| Proportion of heating to grate surface, | 35·033 to 1·000 |
| Proportion of grate surface to absolute calorimeter between tubes, | 5·354 to 1·000 |
| Proportion of grate surface to calorimeter for direct draught, | 6·327 to 1·000 |
| Proportion of grate surface to cross area of smoke-pipe, | 6·731 to 1·000 |
| Capacity of steam room in the four boilers, | 2,980 cubic feet. |
| Weight of water in the four boilers, | 184,700 pounds. |

## FINISHED WEIGHT OF BOILERS, &c.

| | | |
|---|---|---|
| Plate iron, | 209,410 | pounds. |
| Cast iron, | 12,000 | " |
| Brass tubes, | 33,896 | " |
| Smoke-pipe, complete, | 23,978 | " |
| Grate bars, valve-chests, and shells of heaters, all of cast iron, | 27,711 | " |
| Valves, cocks, hand-pump, tubes and tube plates of heaters, &c., all of brass, | 8,531 | " |
| Total, | 315,526 | pounds. |

## SUMMARY OF FINISHED WEIGHTS OF MACHINERY.

| | | |
|---|---|---|
| Engines and dependencies, | 470,118 | pounds. |
| Screw and hoisting apparatus, | 34,460 | " |
| Boilers and dependencies, | 315,526 | " |
| Hoisting engines and gear, | 5,315 | " |
| Auxiliary boiler and dependencies, | 6,137 | " |
| Outfit of ash and coal buckets, gauges, tanks, instruments, &c., | 3,430 | " |
| Duplicate pieces and tools, | 18,463 | " |
| Total weight of metal, | 853,449 | " |
| Paint; felt for boilers, &c.; wood casing; lead and gum in joints, &c., | 14,651 | " |
| Water in boilers, | 184,700 | " |
| Total weight in engine department, exclusive of coal, | 1,052,800 | pounds. |
| Or, | 470 | tons. |
| Weight of coal carried in bunkers, | 620 | " |
| Total weight in engine department, | 1,090 | tons. |

The above weight of metal was made up of the following weights of the kinds enumerated, namely:—

| | | |
|---|---|---|
| Cast iron, | 292,646 | pounds. |
| Wrought iron, | 125,873 | " |
| Plate iron, | 292,030 | " |
| Brass, | 127,000 | " |
| Copper, | 15,000 | " |
| Steel, | 900 | " |
| Total, | 853,449 | pounds. |

## MAXIMUM PERFORMANCE IN SMOOTH WATER, UNINFLUENCED BY WIND OR CURRENT.

The following is the maximum performance that, uninfluenced by wind or current, can be permanently sustained in smooth water with the first quality of steam coal.

The different pressures in the cylinder are the mean of a collation of a large number of indicator diagrams.

| | |
|---|---|
| Vessel's mean draught of water, in feet, | 21·83 |
| Vessel's greatest immersed transverse section, in square feet, | 808·2 |
| Vessel's displacement, in tons, | 4,400·4 |
| Vessel's speed per hour in geographical miles of 6086 feet, | 9·11 |
| Number of double strokes of engines' pistons, and of revolutions of the screw, made per minute, | 49·3 |
| Slip of the screw in per centum of its speed, | 18·5 |
| Portion of the stroke of the piston from the commencement at which the steam is cut off, | $\frac{11}{36}$ |
| Proportion of throttle-valve open, | Wide. |
| Steam pressure in the boilers, in pounds per square inch above the atmosphere, | 15·0 |
| Vacuum in the condenser, in inches of mercury, | 26·0 |
| Steam pressure in the cylinders, in pounds per square inch above zero, at the commencement of the stroke of the piston, | 27·0 |
| Steam pressure in the cylinders, in pounds per square inch above zero, at the point of cutting off, | 25·0 |
| Steam pressure in the cylinders, in pounds per square inch above zero, at the end of the stroke of the piston, | 9·0 |
| Mean back pressure in the cylinders, in pounds per square inch above zero, against the piston during its stroke, | 3·5 |
| Mean gross effective pressure on the piston during its stroke, in pounds per square inch, | 14·3 |
| Mean total pressure on the piston during its stroke, in pounds per square inch, | 17·8 |
| Gross effective horses power developed by the engines, | 1,038·801 |
| Total horses power developed by the engines, | 1,293·054 |
| Temperature of feed-water in degrees Fahr., | 135· |
| Pounds of first quality steam coal consumed per hour, | 4,250· |
| Per centum of refuse from the coal, in ashes, &c., | 12·5 |
| Pounds of coal consumed per hour per square foot of grate surface, | 12·561 |
| Pounds of combustible consumed per hour per square foot of grate surface, | 10·991 |
| Pounds of coal consumed per hour per gross effective indicated horse power, | 4·091 |
| Pounds of coal consumed per hour per total indicated horse power, | 3·287 |
| Pounds of combustible consumed per hour per gross effective indicated horse power, | 3·580 |
| Pounds of combustible consumed per hour per total indicated horse power, | 2·875 |
| Pounds of steam discharged per hour from cylinders into condensers, calculated from the pressure of the steam at the end of the stroke of the piston, | 25,070·461 |

| | |
|---|---|
| Pounds of steam per hour equivalent to the heat annihilated in the cylinders to produce the total power of the engines, calculated from Joule's equivalent, | 3,375·451 |
| Pounds of steam that would have been evaporated per hour, had the heat been so applied that was expended in "blowing off" to maintain the sea water in the boilers at one and three-fourths time the natural concentration, supposing the boilers to evaporate 10 pounds of water per pound of coal, | 5,386·747 |
| Sum of the above three quantities, | 33,832·659 |
| Pounds of water evaporated from temperature of feed-water (135° Fahr.) per hour, supposing 10 pounds of water vaporized by one pound of coal, | 42,500·000 |
| Per centum of the steam evaporated in the boilers not accounted for by the indicator and by "blowing off," being the per centum which the difference between the quantities on the two preceding lines is of the quantity on the preceding line, | 20·39 |

## DISTRIBUTION OF THE POWER DURING THE ABOVE PERFORMANCE.

The pressure required to work the engines and shafting *per se* being taken at 1½ pound per square inch of piston, the power thus absorbed is 108·97 horses.

Deducting from the gross effective power of 1,038·80 horses developed by the engines, this power of 108·97 horses, there remains the power of 929·83 horses applied to the shaft, of which 7½ per centum or 69·74 horses is absorbed by the friction of the load.

The power expended in overcoming the cohesive resistance of the water by the screw blades, calculated in the ratio of the square of the velocity, and for a value of 0·45 pound avoirdupois per square foot of helicoidal surface moving in its helical path with a velocity of 10 feet per second, amounts to 76·72 horses.

The powers (69·74 and 76·72 horses) absorbed by the friction of the load and expended in overcoming the cohesive resistance of the water by the screw blades, being deducted from the power (929·83 horses) applied to the shaft, there remains 783·37 horses power expended in the slip of the screw and in the propulsion of the hull. And as the slip of the screw is 18·5 per centum of its speed, the power expended in it is (783·37×·185=) 144·92 horses, leaving (783·37—144·92=) 638·45 horses expended in the propulsion of the simple hull.

Collecting the foregoing, we have the following distribution of the power; namely:—

| | Horses power. | Per centum. |
|---|---|---|
| Gross effective indicator power developed by the engines, | 1,038·80 | |
| Power required to work the engines and shafting *per se*, | 108·97 | |
| Net power applied to the shaft, | 929·88 | or 100·00 |
| Power absorbed by the friction of the load, | 69·74 | 7·50 |
| Power expended in overcoming the cohesive resistance of the water by the screw blades, | 76·72 | 8·25 |
| Power expended in the slip of the screw, | 144·92 | 15·59 |
| Power expended in the propulsion of the vessel, | 638·45 | 68·66 |
| Totals, | 929·83 | or 100·00 |

## THRUST OF THE SCREW.

The thrust of the screw during the foregoing maximum performance can be calculated from the data in the above distribution of the power. The power required to propel the simple hull is therein found to be 638·45 horses, equal to $(638{\cdot}45 \times 33000 =)$ 21,068,850 pounds raised one foot high per minute. The speed of the vessel was 9·11 geographical miles of 6086 feet per hour or $\left(\frac{9{\cdot}11 \times 6086}{60} =\right)$ 924·0577 feet per minute; the resistance of the vessel at this speed, or its equivalent the thrust of the screw, was consequently $\left(\frac{21068850}{924{\cdot}0577} =\right)$ 22,800·362 pounds.

## PERFORMANCE AT SEA UNDER THE CONDITIONS OF ORDINARY PRACTICE.

In the three following tables will be found abtracts of the steam log of the "WABASH." The first gives the performance under steam alone; the second the performance under steam assisted by the fore and aft sails; the third the performance under steam and square sails combined. These tables include the whole of the performance up to the present date during which all the elements are recorded in the vessel's log. The data of that log were ascertained in the manner described under the head of the "MERRIMACK." Where, in the table of the performance under steam and square sails combined, the minus (—) sign is prefixed to the slip of the screw, it signifies that the speed of the screw (product of pitch and revolutions) was the recorded per centum less than the speed of the vessel, the latter being taken as unity.

The three tables above referred to contain the performance of the vessel in detail under the conditions of wind, sea, and sail stated; the means of the quantities in these tables will be found in the table which follows them under the heading of "Synopsis of the Steam Log of the U. S. Screw Frigate 'WABASH,'" together with the calculated results therefrom, made in the same manner as for the U. S. Screw Frigate "MERRIMACK."

ABSTRACT OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "WABASH," EMBRACING ALL HER PERFORMANCE UNDER STEAM ALONE.

| DATE | Number of consecutive hours. | Course of the Vessel. | WIND. | | STATE OF THE SEA. | Speed of the Vessel per hour, in geographical miles of 6086 feet. | Number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute. | Steam pressure in Boilers, in pounds per sq. inch above the atmosphere. | Vacuum in Condensers, in inches of mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Coal consumed per hour. | Per centum of the Coal lost, in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Direction. | Kind. | | | | | | | | | |
| 1856. | | | | | | | | | | | | | |
| Sept. 10, | 12 | . . | Ahead. | Light breeze. | Mod. swell. | 6·000 | 35·80 | 15·0 | 22·0 | 0·19 | 25·05 | 2948 | 17⅓ |
| Oct 10, 11, & 12, | 51 | . . | Variable. | " " | Smooth. | 7·108 | 42·73 | 12·6 | 28·3 | 0·77 | 26·65 | 3326 | 17 1/10 |
| " 27, | 24 | . . | " | " airs. | " | 8·200 | 43·25 | 14·3 | 24·0 | 1·00 | 16·39 | 3514 | 13⅓ |
| Nov. 28 & 29, | 21 | . . | On bow. | " breeze. | Gentle swell. | 7·810 | 44·03 | 14·0 | 24·6 | 1·00 | 21·78 | 3833 | " |
| Dec. 1, | 24 | . . | Ahead. | " airs. | Smooth. | 6·667 | 36·00 | 10·5 | 24·0 | 1·00 | 18·33 | 3083 | " |
| " 2 & 3, | 34 | . . | " | Gentle breeze. | Mod. swell. | 6·689 | 38·53 | 13·9 | 24·7 | 1·00 | 30·31 | 3448 | " |
| " 5, | 13 | . . | On bow. | Light " | Smooth. | 7·461 | 41·60 | 13·3 | 25·0 | 1·00 | 20·90 | 3255 | " |
| " 27 & 28, | 27 | . . | Ahead. | Gentle " | Mod. swell. | 6·018 | 39·10 | 14·6 | 24·5 | 1·00 | 32·15 | 3400 | 15 |

(TABLE CONTINUED.)

| DATE. | Number of consecutive hours. | Course of the Vessel. | WIND. Direction. | WIND. Kind. | STATE OF THE SEA. | Vessel's mean draught of water. | Speed of the Vessel per hour, in geographical miles of 6086 feet. | Number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute. | Steam pressure in Boilers, in pounds per sq. inch above the atmosphere. | Vacuum in Condensers, in inches of mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Coal consumed per hour. | Per centum of the Coal lost, in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1858. | | | | | | | | | | | | | | |
| Jan. 8, | 4 | N. E. | N. E. | Mod. breeze. | Rough. | | 4·750 | 39·95 | 12·0 | 27·0 | 0·75 | 47·56 | 3500 | 19 3/5 |
| " 14, | 18 | E. | E. | Gentle " | Mod. swell. | | 5·900 | 41·83 | 13·0 | 27·0 | 0·38 | 37·79 | 3100 | " |
| " 16, | 8 | N. | N. by W. | " " | " " | | 5·625 | 40·12 | 13·0 | 27·0 | 0·75 | 38·17 | 2900 | " |
| Feb. 9, | 4 | N. E. | N. N. E. | " " | " " | | 6·375 | 42·25 | 13·0 | 27·0 | 0·25 | 33·46 | 3240 | 14 2/5 |
| " 16, | 24 | N. | N. W. | Fresh " | " " | | 6·000 | 44·32 | 14·0 | 27·0 | 0·75 | 40·30 | 3542 | " |
| June 1, | 11 | S. E. | E. S. E. | Light " | " " | 23·1 | 6·477 | 43·75 | 14·2 | 24·5 | 0·75 | 34·71 | 3230 | 18 |
| " 2, | 12 | S. E. | N. E. | " " | Gentle " | | 7·318 | 45·00 | 15·0 | 24·0 | 0·38 | 28·33 | 3283 | " |
| " 7, | 20 | S. E. | S. E. | Mod. " | Mod. " | | 5·233 | 41·15 | 12·1 | 25·0 | 0·50 | 43·92 | 3064 | " |
| " 15, | 11 | N. W. | N. W. | " " | " " | | 6·227 | 44·27 | 13·5 | 24·5 | 0·50 | 37·96 | 2687 | " |
| " 17, | 24 | . . | On bow. | Light airs. | Smooth. | | 7·250 | 44·42 | 11·8 | 25·0 | 0·50 | 28·04 | 2763 | " |
| July 22 & 23, | 19 | E. N. E. | E. | Gentle breeze. | " | | 6·842 | 42·72 | 12·5 | 25·8 | 0·85 | 34·53 | 2770 | 15 1/5 |
| " 25, | 8 | E. N. E. | N. E. | Fresh " | Heavy swell. | | 3·875 | 39·30 | 12·0 | 26·0 | 1·00 | 56·51 | 3386 | " |
| " 26, | 9 | E. N. E. | E. N. E. | " " | " " | | 3·400 | 38·77 | 12·4 | 26·0 | 0·88 | 61·32 | 3838 | " |
| " 27, | 6 | N. E. by E. | N. | Light " | Mod. " | | 6·292 | 40·83 | 12·0 | 26·0 | 0·50 | 32·04 | 2993 | " |
| Aug. 13, | 24 | E. by S. | E. N. E. | " " | Gentle " | 21·1 | 7·406 | 42·10 | 12·1 | 26·0 | 0·75 | 22·42 | 3534 | 15 4/5 |
| " 16, | 12 | E. S. E. | E. | Gentle " | Smooth. | 21·0 | 7·333 | 43·93 | 12·0 | 26·0 | 1·00 | 26·37 | 3824 | " |
| " 21, | 12 | E. by S. | E. | Light " | " | 22·4 | 7·667 | 43·14 | 11·1 | 26·0 | 0·62 | 21·62 | 3000 | 11 1/2 |
| " 23, | 7 | N. N. E. | N. E. | Strong " | Heavy swell. | | 4·800 | 37·70 | 14·0 | 25·0 | 0·80 | 43·85 | 2750 | " |
| " 24 & 25, | 16 | N. E by E. | N. N. E. | Light " | Rough. | | 5·375 | 40·09 | 12·5 | 25·8 | 0·88 | 40·87 | 3165 | " |
| Sept. 20 & 21, | 16 | S. S. E. | S. | " airs. | Smooth. | 22·2 | 7·562 | 40·61 | 12·0 | 25·7 | 0·50 | 17·88 | 3430 | 11 1/4 |
| " 22, 23, 24, | 47 | S. E by E. | S. | " breeze. | " | 21·9 | 7·644 | 42·25 | 12·8 | 25·5 | 0·47 | 20·21 | 3684 | " |
| " 24, 25 26, | 41 | S. E. by E | S. by E. | Mod. " | Gentle swell. | 21·5 | 6·214 | 41·38 | 12·1 | 25·3 | 0·50 | 33·78 | 3711 | " |
| " 26, | 9 | N. N. E. | N. by E. | Light " | Smooth. | 21·3 | 6·917 | 42·47 | 12·0 | 25·0 | 0·50 | 28·17 | 3306 | " |
| " 28, | 11 | N. E. | N. N. E. | Gentle " | " | 21·0 | 6·368 | 41·50 | 12·4 | 25·0 | 0·62 | 32·38 | 3900 | " |
| Oct. 9, | 8 | N. | N. by W. | Light airs. | " | 22·1 | 8·000 | 44·26 | 13·0 | 25·0 | 0·75 | 20·30 | 3325 | " |
| " 22, | 9 | S. E. | S. by E. | " " | " | 21·10 | 7·861 | 42·93 | 11·7 | 25·0 | 0·44 | 19·24 | 3280 | 16 3/8 |
| " 23 & 24, | 14 | S. E. by E. | Variable. | " breeze. | " | 21·8 | 7·518 | 42·03 | 11·7 | 25·0 | 0·44 | 21·13 | 2934 | " |
| " 25, | 9 | N. E. | N. by E. | " " | " | 21·6 | 7·167 | 41·43 | 11·4 | 24·8 | 0·88 | 23·69 | 3200 | " |
| Nov. 10 & 11, | 12 | W. | . . | Calm. | " | | 7·667 | 44·67 | 14·1 | 24·3 | 0·55 | 24·30 | 3283 | 15 |
| " 11, | 7 | W. | On bow. | Light breeze. | " | | 6·393 | 41·07 | 14·0 | 24·0 | 0·50 | 31·35 | 3454 | " |
| " 12, | 10 | . . | On bow. | Mod. " | Mod. swell. | 21·7 | 4·300 | 39·10 | 14·4 | 24·0 | 0·63 | 51·52 | 3585 | 18 |
| " 18, 19, 20, | 44 | N. N. W. | N. by W. | Light " | " " | 21·8 | 6·881 | 43·00 | 14·4 | 24·1 | 0·65 | 29·43 | 3918 | " |
| " 21 & 22, | 30 | N. W. by W. | N. by W. | Mod. " | " " | | 5·692 | 40·11 | 14·2 | 25·8 | 0·50 | 37·43 | 3950 | " |
| " 28 & 29, | 16 | N. W. by W. | N. by W. | Fresh " | " " | | 5·375 | 40·68 | 14·7 | 24·8 | 0·50 | 41·73 | 3941 | 10 1/4 |
| Dec. 1, | 8 | N. by E. | S. by W. | " " | Smooth. | 21·1 | 8·625 | 47·00 | 14·3 | 25·0 | 0·63 | 19·07 | 4188 | " |
| 1859. | | | | | | | | | | | | | | |
| Feb. 16, | 4 | S. E. | N. | Light breeze | " | 22·2 | 7·563 | 43·25 | 13·8 | 22·0 | 1·00 | 22·88 | 3833 | 15 |
| April 3, | 4 | N. by W. | N. N. W. | Gentle " | " | 22·2 | 6·500 | 43·50 | 12·0 | 25·0 | 0 75 | 34·10 | 4385 | 16 3/5 |
| " 27 & 28, | 34 | S S. E. | . . | Calm. | " | | 7·022 | 43·78 | 15·3 | 25·0 | 0·35 | 19·25 | 3687 | " |
| May 19 & 20, | 19 | N. W. | N. W. | Mod. breeze. | Mod. swell | | 5·787 | 43·10 | 15·2 | 25·0 | 0·50 | 41·80 | 3697 | 11 3/5 |
| June 8 & 9, | 14 | E. by S. | Variable. | Light " | Smooth. | | 7·214 | 44·03 | 15·1 | 22·4 | 0·37 | 27·75 | 3534 | 17 |
| " 13 & 14, | 26 | N. N. W. | N. N. W. | Mod. " | Rough. | | 4·856 | 39·92 | 14·7 | 22·5 | 0·46 | 46·35 | 3750 | 16 |
| " 14 & 15, | 25 | N. W. | N. W. | Light airs. | Smooth. | | 7 870 | 44·08 | 15·0 | 22·2 | 0·40 | 21·25 | 4109 | " |
| July 2, 3, & 4, | 35 | S. S. E. | S. by W. | " breeze. | " | | 7·436 | 41·94 | 12·6 | 23·1 | 0·47 | 21·81 | 3562 | 19 3/5 |
| " 6, | 5 | . . | . . | Calm. | " | | 7·850 | 43·52 | 13·0 | 23·5 | 0·44 | 20·45 | 3563 | " |
| " 10, | 15 | N. | N. by W. | Light airs. | " | | 7·333 | 43·29 | 13·9 | 23·5 | 0·50 | 25·30 | 3166 | " |
| Aug. 1, 2, & 3, | 32 | N. W. | N. by W. | " breeze. | " | | 7·407 | 42·91 | 14·6 | 24·1 | 0·50 | 23·87 | 3670 | 21 3/4 |
| " 26, | 5 | N. W. | N. by W. | " airs. | " | | 7·900 | 45·00 | 15·0 | 24·0 | 0·50 | 22·58 | 3575 | 20 |
| Sept. 27 & 28, | 19 | W. S. W. | S. W. | " " | " | | 7·518 | 43·87 | 14·5 | 24·7 | 0·50 | 24·49 | 3219 | 21 3/5 |
| " 28, | 3 | N. W. | S. by E. | Mod. breeze. | " | | 7·667 | 45·00 | 15·0 | 25·0 | 0·50 | 24·86 | 4150 | " |
| Oct. 5, 6, 7, 8, | 62 | S. by. E. | S. | Light " | " | | 6·787 | 44·47 | 15·1 | 23·1 | 0·50 | 32·70 | 3797 | 18 |
| Nov. 10 & 11, | 27 | . . | . . | " airs. | " | | 7·769 | 44·91 | 14·0 | 23·5 | 0·44 | 23·70 | 3182 | 19 3/8 |
| | | | 22° from ahead. | Gentle breeze. | Gentle swell. | | 6·724 | 42·07 | 13·4 | 24·7 | 0·61 | 29·51 | 3474 | 15 1/10 |

## ABSTRACT OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "WABASH," EMBRACING ALL HER PERFORMANCE UNDER STEAM AND FORE AND AFT SAILS.

| DATE. | Number of consecutive hours. | Course of the Vessel. | WIND. Direction. | WIND. Kind. | STATE OF THE SEA. | Vessel's mean draught of water, in feet and inches. | Speed of the Vessel per hour in geographical miles of 6086 feet. | Number of double strokes of engines' pistons, and of revolutions of the screw, made per minute. | Steam pressure in Boilers, in pounds per sq. inch above the atmosphere. | Vacuum in Condensers, in inches of Mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Anthracite consumed per hour. | Per centum of the Coal lost, in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1856. | | | | | | ft. in. | | | | | | | | |
| Nov. 80, | 24 | . . | On bow. | Strong breeze. | Heavy swell. | | 6·667 | 83·92 | 12·8 | 25·0 | 1·00 | 18·81 | 2052 | 18⅕ |
| 1857. | | | | | | | | | | | | | | |
| Apr. 18, 19, | 12 | S. E. | E. | Gentle " | Smooth. | | 7·000 | 38·75 | 18·0 | 26·0 | 0·42 | 20·84 | 2855 | 16 |
| 1858. | | | | | | | | | | | | | | |
| Jan. 9, 10, | 48 | . . | On bow. | Mod. " | Mod. swell. | | 6·568 | 41·46 | 18·8 | 27·0 | 0·56 | 80·19 | 2866 | 19⅗ |
| Feb. 10, 11, | 48 | . . | On bow. | " " | " " | | 6·833 | 43·92 | 12·2 | 27·0 | 0·62 | 31·40 | 3209 | 14⅖ |
| " 12, | 8 | N. N. E. | On bow. | Fresh " | " " | | 8·125 | 45·00 | 11·5 | 27·0 | 0·38 | 20·37 | 3470 | " |
| " 18, | 12 | N. E. | N. N. W. | Mod. " | Rough. | | 6·838 | 44·17 | 18·0 | 27·0 | 0·38 | 31·77 | 3150 | " |
| " 18, 14, | 22 | N. E. | N. by W. | Fresh " | Heavy swell. | | 3·909 | 37·54 | 12·7 | 27·0 | 0·32 | 54·08 | 2864 | " |
| " 15, | 24 | N. by E. | N. W. | Strong " | Rough. | | 5·250 | 42·15 | 13·5 | 27·0 | 1·00 | 45·07 | 819· | " |
| June 6, | 12 | . . | On bow. | Mod. " | Mod. swell. | 22 0 | 7·083 | 44·17 | 12·3 | 25·5 | 1·00 | 29·27 | 3817 | 18 |
| " 7, | 4 | S. E. by S. | E. by N. | Fresh " | " " | | 7·625 | 45·00 | 13·0 | 25·0 | 0·38 | 25·27 | 2600 | " |
| " 16, | 24 | W. N. W. | N. | " " | Smooth. | | 6·870 | 43·25 | 11·0 | 28·5 | 0·88 | 29·95 | 2917 | " |
| July 28, 24, | 16 | N. by W. | W. N. W. | Gentle " | " | | 7·300 | 41·52 | 10·2 | 25·8 | 0·88 | 22·46 | 3004 | 15⅕ |
| " 25, | 4 | E. N. E. | E. S. E. | Mod. " | Rough. | | 5·250 | 41·15 | 11·0 | 26·0 | 0·62 | 43·74 | 3544 | " |
| " 26, 27, | 31 | N. N. E. | E. N. E. | " " | " | | 5·170 | 40·27 | 11·3 | 26·0 | 0·81 | 48·38 | 3220 | " |
| Aug. 12, | 12 | E. by S. | N. E. | Light " | Gentle swell. | 21 3 | 7·800 | 42·10 | 12·3 | 26·0 | 0·50 | 18·29 | 3181 | 15⅘ |
| " 14, 15, 16 | 56 | E. by S. | N. E. | Gentle " | Mod. " | 20 10 | 8·125 | 42·28 | 10·5 | 25·4 | 0·67 | 18·95 | 3365 | " |
| " 22, | 16 | E. S. E. | E. N. E. | Strong " | Rough. | 22 2 | 4·600 | 39·36 | 14·0 | 25·5 | 0·94 | 48·46 | 2885 | 11½ |
| " 22, 23, | 17 | S. E. by E. | N. by E. | " " | Heavy swell. | 22 0 | 2·120 | 24·96 | 10·4 | 28·5 | 0·27 | 62·54 | 1878 | " |
| " 23, 24, | 12 | . . | On bow. | Fresh " | " " | | 3·980 | 37·20 | 12·7 | 25·0 | 0·83 | 52·82 | 2707 | " |
| " 25, | 10 | N. E. by E. | N. | Light " | Mod. " | | 7·100 | 44·88 | 14·5 | 25·0 | 1·00 | 26·84 | 3500 | " |
| " 26, 27, | 29 | N. E. | N. | Strong " | Rough. | | 3·320 | 32·43 | 9·1 | 25·2 | 0·37 | 54·85 | 2808 | " |
| " 28, | 10 | N. E. | N. by W. | Gentle " | " | 22 0 | 5·100 | 33·76 | 11·0 | 24·0 | 0·27 | 38·88 | 2836 | " |
| Sept. 21, | 11 | S. E | S. S. W. | Light " | Smooth. | 22 1 | 8·045 | 42·06 | 11·8 | 25·0 | 0·50 | 15·64 | 3502 | 11¼ |
| " 24, | 4 | S. E. by E. | S. | Gentle " | " | 21 7 | 8·187 | 43·40 | 12·0 | 25·0 | 0·50 | 16·81 | 3945 | " |
| " 27, | 22 | N. E. by E. | N. | Mod. " | Mod. swell. | 21 1 | 5·807 | 40·50 | 12·2 | 25·0 | 0·62 | 36·76 | 3774 | " |
| Oct. 24, | 10 | S. E. by E. | N. E. | Light " | Smooth. | 21 7 | 7·825 | 42·68 | 12·2 | 25·0 | 0·38 | 19·15 | 3230 | 16⅓ |
| " 25, | 7 | S. E. | N. by E. | Gentle " | " | 21 6 | 7·500 | 40·90 | 11·6 | 24·5 | 0·44 | 19·13 | 2854 | " |
| Nov. 4, | 10 | S. | S. E. | Mod. " | " | | 7·600 | 44·12 | 14·0 | 24·0 | 0·50 | 24·04 | 3075 | 20½ |
| " 30, | 24 | N. | On bow. | " " | Gentle swell. | | 7·312 | 43·37 | 14·0 | 25·0 | 0·50 | 25·65 | 4108 | 10¼ |
| Dec. 1, | 12 | N. | Abeam. | Gentle " | Smooth. | | 8·250 | 45·01 | 13·2 | 25·0 | 0·50 | 19·16 | 4308 | " |
| 1859. | | | | | | | | | | | | | | |
| Apr. 26, | 4 | S. | W. S. W. | Light " | " | 21 11 | 6·625 | 41·75 | 15·0 | 26·0 | 0·81 | 30·02 | 3780 | 16⅗ |
| June 6, 7, | 18 | S. S. W. by W. | W. by S. | " " | Gentle swell. | | 7·390 | 43·88 | 15·0 | 25·0 | 0·37 | 25·78 | 3484 | 17 |
| " 12, | 12 | N. E. | S. E | " " | Smooth. | | 7·604 | 42·45 | 14·3 | 21·8 | 0·31 | 21·00 | 3709 | 16 |
| July 5, | 4 | S. W. | W. | Gentle " | Mod. swell. | | 7·250 | 42·20 | 12·0 | 23·5 | 0·31 | 24·28 | 3622 | 19⅗ |
| Oct. 26, | 7 | N W. | N. by E. | " " | Gentle " | | 6·964 | 45·20 | 15·3 | 23·5 | 0·50 | 32·06 | 3718 | 16 |
| Nov. 6, 7, 8, | 53 | W. S. W. | S. S. W. | Light " | " " | | 6·948 | 43·82 | 14·4 | 23·3 | 0·50 | 30·07 | 3100 | 19⅔ |
| " 9, | 10 | S. W. | S. | Gentle " | Smooth. | | 7·650 | 43·77 | 15·0 | 23·5 | 0·44 | 22·92 | 2799 | " |
| " 13, | 4 | W. by N. | N. by W. | Light " | Mod. swell. | | 5·750 | 35·15 | 12·5 | 23·0 | 0·31 | 27·86 | 1925 | 17⅓ |
| Dec. 4, | 8 | W. by N. | S. | " " | " " | | 7·438 | 40·32 | 13·0 | 22·0 | 0·50 | 18·64 | 2642 | 19⅗ |
| " 7, | 8 | N. W. by W. | N. by E. | Gentle " | " " | | 7·812 | 43·61 | 14·0 | 21·0 | 0·50 | 21·00 | 3175 | " |
| " 13, | 17 | N. | N. E. | Strong " | " " | | 7·044 | 46·06 | 15·0 | 28·0 | 0·50 | 32·53 | 4104 | " |
| " 15, | 8 | N. by E. | N. W. | Mod. " | " " | | 7·750 | 48·22 | 14·0 | 24·5 | 0·50 | 29·12 | 3825 | " |
| " 16, | 7 | N. by W. | N. E. by N. | Gentle " | Gentle " | | 7·572 | 46·86 | 14·0 | 28·5 | 0·50 | 28·78 | 3843 | " |
| | | | 56° from Ahead. | Mod. breeze. | Mod. swell. | | 6·526 | 41·22 | 12·6 | 25·1 | 0·59 | 30·18 | 3156 | 15½ |

## ABSTRACT OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "WABASH," EMBRACING ALL HER PERFORMANCE UNDER STEAM AND SQUARE SAILS COMBINED.

| DATE. | Number of consecutive hours. | Course of the Vessel. | WIND. Direction. | WIND. Kind. | STATE OF THE SEA. | Vessel's mean draught of water, in feet and inches. | Speed of the Vessel per hour, in geographical miles of 6086 feet. | Number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute. | Steam pressure in Boilers, in pounds per sq. inch above the atmosphere. | Vacuum in Condensers, in inches of mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Coal consumed per hour. | Per centum of the Coal lost, in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1856. | | | | | | ft. in. | | | | | | | | |
| Oct. 1 & 2, | 24 | . . | On Quarter. | Light breeze. | Smooth. | | 9·000 | 45·82 | 13·2 | 27·7 | 0·60 | 12·41 | 3124 | $19\frac{1}{3}$ |
| " 9 & 10, | 16 | . . | " | " " | " | | 10·000 | 47·90 | 14·2 | 28·0 | 0·75 | 7·92 | 3717 | $17\frac{1}{10}$ |
| Nov. 28, | 3 | . . | " | " " | " | | 9·167 | 44·50 | 14·0 | 24·0 | 1·00 | 8·16 | 2817 | $13\frac{1}{5}$ |
| 1858. | | | | | | | | | | | | | | |
| Jan. 8, | 8 | N. W. | E. | Mod. " | Mod. swell. | | 8·500 | 42·00 | 13·0 | 27·0 | 0·25 | 10·75 | 2500 | $19\frac{3}{5}$ |
| Feb. 2, | 7 | S. S. W. | N. | " " | Smooth. | | 7·430 | 41·40 | 13·0 | 27·0 | 0·33 | 20·85 | 2397 | $14\frac{2}{5}$ |
| " 12, | 9 | E. N. E. | N. | " " | Mod. swell. | | 6·333 | 44·00 | 13·0 | 27·0 | 0·75 | 36·52 | 3490 | " |
| " 14, | 14 | N. E. | N. by W. | Strong " | Heavy " | | 7·571 | 41·48 | 12·5 | 27·0 | 0·25 | 19·39 | 2580 | " |
| June 2, | 8 | S. | N. | Fresh " | Mod. swell. | | 8·000 | 44·50 | 15·0 | 25·5 | 0·37 | 20·71 | 2663 | 18 |
| " 3 & 4, | 34 | S. | N. N. E. | Mod. " | " " | | 8·200 | 39·88 | 10·0 | 25·4 | 0·86 | 9·32 | 2730 | " |
| " 8, 9, 10, | 55 | . . | On Quarter. | " " | " " | 21 11 | 8·473 | 43·56 | 10·7 | 25·0 | 0·47 | 14·22 | 2611 | " |
| " 13, | 8 | W. | E. S. E. | Fresh " | " " | | 7·970 | 45·00 | 11·0 | 25·0 | 0·50 | 21·90 | 2600 | " |
| July 24, | 4 | N. | W. | Mod. " | Smooth. | | 10·500 | 45·80 | 11·2 | 26·0 | 0·56 | –1·10 | 3045 | $15\frac{1}{5}$ |
| " 28, | 4 | N. E. by E. | W. | Light | Gentle swell. | | 6·875 | 42·12 | 12·0 | 26·0 | 0·50 | 28·02 | 2756 | " |
| Aug. 25 & 26, | 10 | N. E. by E. | S. by W. | Gentle " | Smooth. | | 9·675 | 46·46 | 18·7 | 25·8 | 0·68 | 8·16 | 3050 | $11\frac{1}{2}$ |
| " 27, | 5 | N. E. by E. | N. by W. | Fresh " | " | | 9·400 | 40·96 | 10·5 | 25·0 | 0·25 | –1·20 | 2920 | " |
| Sept. 1 & 2, | 13 | . . | On Quarter. | Mod. " | " | 21 9 | 9·577 | 38·45 | 12·5 | 26·0 | 0·37 | –8·97 | 2785 | $11\frac{1}{3}$ |
| " 24, | 3 | S. E. by E. | S. by W. | " " | " | | 8·333 | 44·43 | 12·0 | 25·0 | 0·25 | 17·29 | 3107 | $11\frac{1}{4}$ |
| Oct. 20, 21, 22 | 21 | . . | On Quarter. | Gentle " | " | 21 11 | 8·464 | 41·76 | 11·3 | 25·0 | 0·32 | 10·62 | 2534 | $16\frac{1}{3}$ |
| " 24, | 4 | S. E. by E. | N. by E. | Light | " | 21 7 | 7·375 | 38·25 | 12·0 | 24·0 | 0·25 | 14·97 | 2575 | " |
| Nov. 4, | 4 | S. W. by S. | S. by E. | " " | " | | 9·000 | 46·65 | 15·0 | 24·0 | 0·50 | 14·92 | 3350 | $20\frac{1}{2}$ |
| " 11, | 4 | W. | S. | Gentle " | " | | 9·000 | 44·15 | 12·0 | 24·0 | 0·50 | 10·10 | 3825 | 15 |
| " 20, | 18 | N. W. by W. | S. by W. | " " | " | | 8·820 | 46·02 | 14·2 | 25·0 | 0·50 | 15·50 | 3990 | 13 |
| " 21, | 7 | N. W. by W. | S | Fresh | Mod. swell. | | 10·250 | 47·06 | 14·6 | 25·0 | 0·44 | 3·95 | 3600 | " |
| " 29, | 10 | N. W. by W. | N. | " " | " " | | 7·950 | 44·98 | 14·6 | 25·0 | 0·50 | 22·05 | 4280 | $10\frac{1}{4}$ |
| 1859. | | | | | | | | | | | | | | |
| May 24 & 25, | 14 | S. E. | N. by W. | Gentle " | Smooth. | | 8·000 | 44·23 | 15·0 | 25·0 | 0·37 | 19·94 | 3139 | 13 |
| July 4, | 17 | S. E. | N. by W. | " " | " | | 8·353 | 42·12 | 12·0 | 28·5 | 0·34 | 12·54 | 3405 | $19\frac{3}{5}$ |
| " 5, | 16 | S. W. by W. | N. W. | " " | " | | 9·220 | 42·82 | 12·8 | 28·5 | 0·37 | 5·00 | 3623 | " |
| Oct. 26, | 13 | N. N. W. | S. E. | " " | Mod. swell. | | 8·854 | 46·08 | 14·8 | 23·5 | 0·50 | 15·27 | 3621 | 16 |
| Nov. 9, | 10 | W. S. W. | N. | " " | Smooth. | | 8·056 | 44·22 | 14·0 | 23·5 | 0·31 | 19·71 | 2950 | $19\frac{2}{3}$ |
| " 13, | 4 | W. by N. | E. | Mod. " | Mod. swell. | | 8·625 | 45·00 | 15·0 | 24·0 | 0·81 | 15·47 | 2425 | $17\frac{1}{3}$ |
| Dec. 3, | 11 | W. by N. | S. by E. | Gentle " | Gentle " | | 8·600 | 42·22 | 12·8 | 23·0 | 0·37 | 14·25 | 2427 | $19\frac{2}{3}$ |
| " 6, | 11 | N. W. by W. | S. by E. | " " | Mod. " | | 7·636 | 43·12 | 14·0 | 23·0 | 0·50 | 21·90 | 2897 | " |
| " 8, | 4 | N. W. by W. | S. by E. | Mod. " | " " | | 8·125 | 44·00 | 13·0 | 21·0 | 0·50 | 18·56 | 3150 | " |
| " 12, | 13 | . . | Abeam. | Strong " | Rough. | | 8·600 | 43·85 | 13·6 | 22·0 | 0·37 | 13·51 | 3502 | " |
| | | | 125° from Ahead. | Mod. breeze. | Gentle swell. | | 8·551 | 43·47 | 12·6 | 25·1 | 0·44 | 13·25 | 3035 | $16\frac{4}{5}$ |

## SYNOPSIS OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "WABASH."

| | | Performance under Steam alone. | Performance under Steam and the Fore and Aft Sails. | Performance under Steam and the Sq. Sails combined. | Mean of the three preceding columns. |
|---|---|---|---|---|---|
| | Total number of hours, | 1,079· | 711· | 406. | 2,196· |
| | Kind of wind, | Gentle breeze. | Mod'ate breeze. | Mod'ate breeze. | . . |
| | Angle made from ahead by the wind with the line of the vessel's keel, | 22° | 56° | 125° | 52° |
| | State of the sea, | Gentle swell. | Moderate swell. | Gentle swell. | . . |
| | Vessel's draught of Water, in feet and inches, Forward, | 20·11 | 20·11 | 20·11 | 20·11 |
| | Vessel's draught of Water, in feet and inches, Mean, | 21·10 | 21·10 | 21·10 | 21·10 |
| | Vessel's draught of Water, in feet and inches, Aft, | 22·9 | 22·9 | 22·9 | 22·9 |
| | Vessel's greatest immersed Transverse Section, in square feet, | 808·2 | 808·2 | 808·2 | 808·2 |
| | Vessel's Displacement, in tons, | 4,400·4 | 4,400·4 | 4,400·4 | 4,400·4 |
| | Mean Speed of the vessel per hour, in geographical miles of 6086 feet, | 6·724 | 6·526 | 8·551 | 6·998 |
| ENGINES. | Mean number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute, per counter, | 42·07 | 41·22 | 43·47 | 42·05 |
| | Mean steam pressure in the Boilers, in pounds per square inch above the atmosphere, | 13·4 | 12·6 | 12·6 | 13·0 |
| | Mean gross effective pressure on Pistons, in pounds per square inch, per indicator, | . . | . . | . . | 10·9 |
| | Mean total pressure on Pistons, in pounds per square inch, per indicator, | . . | . . | . . | 16·1 |
| | Gross effective horses power developed by the Engines, per indicator, | . . | . . | . . | 675·370 |
| | Total horses power developed by the Engines, per indicator, | . . | . . | . . | 997·566 |
| | Vacuum in Condensers, in inches of mercury per gauge, | 24·7 | 25·1 | 25·1 | 24·9 |
| | Steam cut off at in Cylinders from commencement of stroke of piston, | $\frac{11}{36}$ | $\frac{11}{36}$ | $\frac{11}{36}$ | $\frac{11}{36}$ |
| | Proportion of Throttle-valve open, | 0·61 | 0·59 | 0·44 | 0·57 |
| | Slip of the Screw, in per centum of its speed, | 29·51 | 30·18 | 13·25 | 26·61 |
| | Temperature of the feed-water entering the Boilers, in degrees Fahr., | 135· | 135· | 135· | 135· |
| COAL. | Number of pounds of Coal consumed per hour, | 3,474· | 3,156· | 3,035· | 3290· |
| | Per centum of waste of Coal in ashes, clinker, and fine coal, | $15\frac{1}{10}$ | $15\frac{1}{2}$ | $16\frac{2}{5}$ | $15\frac{1}{2}$ |
| | Number of pounds of Combustible consumed per hour, | 2,949· | 2,667· | 2,537· | 2,780· |
| | Number of pounds of Coal consumed per hour, per square foot of grates, | 10·268 | 9·328 | 9·030 | 9·724 |
| | Number of pounds of Combustible consumed per hour per square foot of grates, | 8·716 | 7·883 | 7·498 | 8·217 |
| | Number of pounds of Coal consumed per hour per gross effective indicated horse power, | . . | . . | . . | 4·871 |
| | Number of pounds of Coal consumed per hour per total indicated horse power, | . . | . . | . . | 3·298 |
| | Number of pounds of Combustible consumed per gross effective indicated horse power, | . . | . . | . . | 4·116 |
| | Number of pounds of Combustible consumed per hour per total indicated horse power, | . . | . . | . . | 2·787 |
| ANALYSIS OF ALL INDICATOR DIAGRAMS. | Mean steam pressure in Cylinders at commencement of stroke of piston, in pounds per sq. inch above zero, | . . | . . | . . | 24·5 |
| | Mean steam pressure in Cylinders at the point of cutting off, in pounds per square inch above zero, | . . | . . | . . | 22·5 |
| | Mean steam pressure in Cylinders at the end of the stroke of the piston, in pounds per square inch above zero, | . . | . . | . . | 7·8 |
| | Mean back pressure in Cylinders against the pistons, in pounds per square inch above zero, | . . | . . | . . | 5·2 |
| EVAPORATION FROM FEED-WATER TEMPERATURE 135° FAHR. | Pounds of Steam discharged per hour from the Cylinders, calculated from the pressure at end of stroke of piston, | . . | . . | . . | 18,663·467 |
| | Pounds of Steam condensed per hour in the Cylinders to produce the power developed by the Engines, calculated from JOULE's equivalent, | . . | . . | . . | 2,591·120 |
| | Pounds of Steam that would have been evaporated per hour, had the heat been applied to evaporation that was lost by "blowing off," to maintain the sea water in the boilers at $1\frac{2}{3}$ time the natural concentration, supposing the boilers to evaporate $9\frac{1}{2}$ pounds of water per pound of anthracite, | . . | . . | . . | 3,484·771 |
| | Sum of the above three quantities, | . . | . . | . . | 25,099·858 |
| | Pounds of Water evaporated from temperature of Feed-water (135° Fahr.) per hour, supposing $9\frac{1}{2}$ pounds of water vaporized by one pound of Anthracite, | . . | . . | . . | 31,255·000 |
| | Per centum of the Steam evaporated in the boilers not accounted for by the indicator, and by "blowing off," being the per centum which the difference between the quantities on the two preceding lines is of the quantity on the preceding line, | . . | . . | . . | 19·70 |

# "MINNESOTA."

## HULL.

In all other respects, except the following dimensions and proportions, the hull of the "MINNESOTA" is the same as that of the "MERRIMACK."

| | |
|---|---|
| Length on mean load-line from forward side of rabbet of stem to after side of rabbet of stern-post, | 264 feet 8½ inches. |
| Extreme breadth on mean load water-line, | 51 " 4 " |
| Moulded breadth on mean load water-line, | 50 " 2 " |
| Depth from mean load water-line to lower edge of rabbet of keel, | 21 " |
| Depth of keel below lower edge of rabbet, | 2 " |
| Depth from top throat timbers to underside of spar-deck plank amidships, | 33 " 8 " |
| Depth from top throat timbers to underside of gun-deck plank amidships, | 26 " 2 " |
| Depth from top throat timbers to underside of berth-deck plank amidships, | 18 " 9 " |
| Height of lower port sill above rabbet of keel amidships, | 30 " |
| Height of upper port sill above rabbet of keel amidships, | 37 " 4 " |
| Displacement to mean load water-line, | 168,839 cubic feet. |
| Displacement to mean load water-line, | 4,833·36 tons. |
| Area of mean load water-line, | 11,318·10 square feet. |
| Displacement per inch of draught at mean load water-line, | 27·00 tons. |
| Area of greatest immersed transverse section to mean load water-line, | 868·10 square feet. |
| Angle of bow on mean load water-line, | 65° |
| Angle of stern on mean load water-line, | 77° |
| Angle of bow on water-line at 9½ feet above lower edge of rabbet of keel, | 42° |
| Angle of stern on water-line at 9½ feet above lower edge of rabbet of keel, | 39° |
| Angle of dead-rise at greatest immersed transverse section, | 14° |
| Ratio of length to breadth on load water-line, | 5·156 |
| Ratio of area of greatest immersed transverse section to area of circumscribing parallelogram, | 0·8053 |
| Ratio of area of load water-line to area of circumscribing parallelogram, | 0·8329 |
| Ratio of displacement to circumscribing parallelopipedon, | 0·5917 |

## ENGINES.

Two condensing, horizontal, trunk engines, constructed acccording to the well known plan of PENN, with the exception that the cut-off valve is a separate slide placed in advance of the steam-valve and working upon a fixed seat. A small pass-over valve, controlled by hand, opens the steam communication between the backs of the cut-off and steam-valves whenever it is desired to use the steam without expansion, or to start the engines. The steam-valve has only lap enough (⅛ inch) to secure the covering of the

port, and the expansion of the steam is produced entirely by the cut-off valve which closes at one-third of the stroke of the piston from the commencement. The steam-valve is a double-ported packed slide worked directly by a Stephenson Link and placed on one side of the cylinder with its face vertical; while the cut-off valve is placed immediately above it and is moved on a horizontal seat by means of a rock-shaft worked by an eccentric. The stems of the steam and cut-off valves are in the same vertical line. This arrangement has the very objectionable feature of leaving a large space between the steam and cut-off valves. The air-pump and feed-pumps are worked directly from the trunk. The following are the dimensions of the engines; namely:—

| | |
|---|---|
| Diameter of the cylinders, | 79½ inches. |
| Diameter of the trunk, | 33 " |
| Area of the annular superficies between the cylinder and trunk, | 4108·62 square inches. |
| Diameter of a plain cylinder having an area equal to the annular superficies between the cylinder and trunk, | 72·326 inches. |
| Stroke of the piston, | 36 " |
| Space displacement of both pistons per stroke, | 171·193 cubic feet. |
| Steam space between the piston at the end of its stroke and the face of the steam-valve, comprising contents of clearance and steam passage at one end of one cylinder, | 4·153 " " |
| Steam space between the faces of the steam and cut-off valves of one cylinder, | 20·700 " " |
| Bulk of steam exhausted from both cylinders at the end of each stroke of the piston, | 179·500 " " |
| Area of steam-port (39 by 6 inches), | 234 square inches. |
| Area of exhaust-port (39 by 8 inches), | 312 " " |
| Area of cut-off valve-port (four openings, each 2¾ by 16½ inches), | 181½ " |
| Diameter of air-pump (double-acting), | 22 inches. |
| Stroke of air-pump piston, | 36 " |
| Space displacement of air-pump piston per stroke, | 7·92 cubic feet. |
| Area of the receiving and of the delivering valves of the air-pump, | 290 square inches. |
| Capacity of the condenser, | 78 cubic feet. |
| Diameter of the feed-pump (single-acting), | 6½ inches. |
| Stroke of the feed-pump plunger, | 36 " |
| Diameter of crank-shaft journals, | one of 14 and two of 13½ " |
| Length of crank-shaft journals, | 25 " |
| Diameter of crank-pin journals, | 13 " |
| Length of crank-pin journals, | 11 " |
| Diameter of trunk-pin journal, | 7½ " |
| Length of trunk-pin journal, | 12 " |
| Diameter of connecting-rod in the neck, | 7½ " |
| Length of connecting-rod between centres, | 8 feet. |
| Length of thrust-bearing, | 30 inches. |
| Number of collars in thrust-bearing, | 7. |
| Acting surface of collars in thrust-bearing, | 590 square inches. |

| | |
|---|---|
| Diameter of injection-valve, | 6 inches. |
| Diameter of outboard delivery-valve, | 18 " |
| Total width of piston, | 7⅞ " |
| Width of packing space of piston, | 5½ " |
| Extreme length occupied by engines in fore and aft direction of vessel, | 19 feet 2 inches. |
| Extreme breadth occupied by engines athwartship the vessel, exclusive of the 18 inches protrusion of trunk, | 22 " 6 " |
| Extreme height occupied by engines, | 9 " |

## FINISHED WEIGHT OF ENGINES.

| | |
|---|---|
| Cast iron, including stairs, floors, deck-plates, &c., | 231,270 pounds. |
| Wrought iron, | 113,040 " |
| Brass, | 54,346 " |
| Steel, | 900 " |
| Copper forgings, | 2,055 " |
| Copper pipes, | 12,411 " |
| Plate iron in coal bunkers, bulkheads, ventilators, and shaft-alley, | 57,617 " |
| Total, | 471,639 pounds. |

## BOILERS.

The boilers are in all respects, the exact duplicates of those of the U. S. Screw Frigate "MERRIMACK" previously described under head of that vessel.

## FINISHED WEIGHT OF BOILERS.

| | |
|---|---|
| Plate iron, | 223,822 pounds. |
| Cast iron, | 5,204 " |
| Brass tubes, | 40,265 " |
| Smoke-pipe complete, | 22,867 " |
| Grate bars, valve-chests, and shells of heaters, all of cast iron, | 24,400 " |
| Tubes and tube plates of heaters, cocks, valves, and other dependencies, all of brass, | 8,676 " |
| Total, | 325,234 pounds. |

## SCREW.

One true screw of uniform length from axis to periphery. It is of bronze and arranged to hoist with the usual lifting apparatus.

| | |
|---|---|
| Diameter, | 17 feet 4 inches. |
| Diameter of hub, | 2 " 1½ " |
| Pitch, | 23 " |
| Length in direction of axis, | 3 " 6 " |

| | |
|---|---|
| Number of blades, | 2. |
| Fraction used of the pitch, | 0·304 |
| Mean angle of blade in function of surface and of propelling efficiency, | 32° 21¼′ |
| Radius of the centre of pressure of the blade, | 6·88 feet. |
| Thickness of the blade above fillet, at the radius of 2 feet, | 8 inches. |
| Helicoidal area of the two blades, | 86·25 square feet. |
| Projected area of the two blades on a plane at right angles to axis, | 69·94 " " |
| Weight of screw, | 22,070 pounds. |
| Weight of hoisting apparatus of screw, including guides, all of brass, | 14,530 " |
| Diameter of forward journal of screw hub, | 20 inches. |
| Length of forward journal of screw hub, | 12¾ " |
| Diameter of after journal of screw hub, | 15 " |
| Length of after journal of screw hub, | 12¾ " |

## SUMMARY OF FINISHED WEIGHTS OF MACHINERY.

| | |
|---|---|
| Engines and dependencies, | 471,639 pounds. |
| Screw and hoisting apparatus, | 36,600 " |
| Boilers and dependencies, | 325,234 " |
| Hoisting engines and gear, | 5,200 " |
| Auxiliary boiler and dependencies, | 5,800 " |
| Outfit of ash and coal buckets, gauges, tanks, instruments, &c., | 3,500 " |
| Duplicate pieces and tools, | 18,600 " |
| Total weight of metal, | 866,573 pounds. |
| Paint, felt for boilers, &c., wood casing, lead and gum in joints, &c., | 14,967 " |
| Water in boilers, | 184,700 " |
| Total weight in engine department, exclusive of coal, | 1,066,240 pounds. |
| Or, | 476 tons. |
| Weight of coal carried in bunkers, | 620 tons. |
| Total weight in engine department, | 1,096 tons. |

The above weight of metal was made up of the following weights of the kinds enumerated, namely:—

| | |
|---|---|
| Cast iron, | 263,800 pounds. |
| Wrought iron, | 129,300 " |
| Plate iron, | 312,500 " |
| Brass, | 145,300 " |
| Copper, | 14,673 " |
| Steel, | 1,000 " |
| Total, | 866,573 pounds. |

## MAXIMUM PERFORMANCE IN SMOOTH WATER, UNINFLUENCED BY WIND OR CURRENT.

The following is the maximum performance that, uninfluenced by wind or current, can be permanently sustained in smooth water with the first quality of steam coal.

The different pressures in the cylinder are the mean of a collation of a large number of indicator diagrams.

| | |
|---|---|
| Vessel's mean draught of water, in feet, | 21·83 |
| Vessel's greatest immersed transverse section, in square feet, | 808·2 |
| Vessel's displacement, in tons, | 4,455·4 |
| Vessel's speed per hour, in geographical miles of 6086 feet, | 8·95 |
| Number of double strokes of engines' pistons, and of revolutions of the screw, made per minute, | 48·3 |
| Slip of the screw, in per centum of its speed, | 18·26 |
| Portion of the stroke of the piston from the commencement, at which the steam is cut off, | ½. |
| Proportion of throttle-valve open, | Wide. |
| Steam pressure in the boilers, in pounds per square inch above the atmosphere, | 11·3 |
| Vacuum in the condenser, in inches of mercury, | 26. |
| Steam pressure in the cylinders, in pounds per square inch above zero, at the commencement of the stroke of the piston, | 23·3 |
| Steam pressure in the cylinders, in pounds per square inch above zero, at the point of cutting off, | 21·3 |
| Steam pressure in the cylinders, in pounds per square inch above zero, at the end of the stroke of the piston, | 8·4 |
| Mean back pressure in the cylinders, in pounds per square inch above zero, against the piston during its stroke, | 3·0 |
| Mean gross effective pressure on the piston during its stroke, in pounds per square inch, | 13·8 |
| Mean total pressure on the piston during its stroke, in pounds per square inch, | 16·8 |
| Gross effective horses power developed by the engines, | 995·840 |
| Total horses power developed by the engines, | 1,212·327 |
| Temperature of feed-water, in degrees Fahrenheit, | 135. |
| Pounds of first quality steam coal consumed per hour, | 4,250. |
| Per centum of refuse from the coal in ashes, &c., | 12·5 |
| Pounds of coal consumed per hour per square foot of grate surface, | 12·561 |
| Pounds of combustible consumed per hour per square foot of grate surface, | 10·991 |
| Pounds of coal consumed per hour per gross effective indicated horse power, | 4·268 |
| Pounds of coal consumed per hour per total indicated horse power, | 3·506 |
| Pounds of combustible consumed per hour per gross effective indicated horse power, | 3·735 |
| Pounds of combustible consumed per hour per total indicated horse power, | 3·067 |
| Pounds of steam discharged per hour from cylinders into condensers, calculated from the pressure of the steam at the end of the stroke of the piston, | 23,169·307 |
| Pounds of steam per hour, equivalent to the heat annihilated in the cylinders to produce the total power of the engines, calculated from Joule's equivalent, | 3,157·391 |

| | |
|---|---|
| Pounds of steam that would have been evaporated per hour, had the heat been so applied that was expended in "blowing off" to maintain the sea-water in the boilers at one and three-fourths time the natural concentration, supposing the boilers to evaporate 10 pounds of water per pound of coal, . . | 5,087·250 |
| Sum of the above three quantities, . . . . . . | 31,413·948 |
| Pounds of water evaporated from temperature of feed-water (135° Fahr.) per hour, supposing 10 pounds of water vaporized by one pound of coal, . . | 42,500· |
| Per centum of the steam evaporated in the boilers not accounted for by the indicator, and by "blowing off," being the per centum which the difference between the quantities on the two preceding lines is of the quantity on the preceding line, | 26·09 |

## DISTRIBUTION OF THE POWER DURING THE ABOVE PERFORMANCE.

The pressure required to work the engines and shafting *per se*, being taken at 1½ pound per square inch of piston, the power thus absorbed is 108·24 horses.

Deducting from the gross effective power of 995·84 horses developed by the engines, this power of 108·24 horses there remains 887·60 horses applied to the shaft, of which 7½ per centum or 66·57 horses is absorbed by the friction of the load.

The power expended in overcoming the cohesive resistance of the water by the screw blades, calculated in the ratio of the square of the velocity, and for a value of 0·45 pound avoirdupois per square foot of helicoidal surface moving in its helical path with a velocity of 10 feet per second amounts to 79·50 horses.

The powers (66·57 and 79·50 horses) absorbed by the friction of the load, and expended in overcoming the cohesive resistance of the water by the screw blades, being deducted from the power (887·60 horses) applied to the shaft, there remains 741·53 horses power expended in the slip of the screw and in the propulsion of the hull. And as the slip of the screw is 18·26 per centum of its speed, the power expended in it is (741·53 × ·1826 =) 135·40 horses, leaving (741·53—135·40 =) 606·13 horses expended in the propulsion of the simple hull.

Collecting the foregoing we have the following distribution of the power; namely:—

| | Horses power. | Per centum. |
|---|---|---|
| Gross effective indicator power developed by the engines, . . . | 995·84 | |
| Power required to work the engines and shafting *per se*, . . . . | 108·24 | |
| Net power applied to the shaft, . . . . . . . | 887·60 | or 100·00 |
| Power absorbed by the friction of the load, . . . . . . | 66·57 | " 7·50 |
| Power expended in overcoming the cohesive resistance of the water by the screw blades, | 79·50 | " 8·96 |
| Power expended in the slip of the screw, . . . . . | 135·40 | " 15·25 |
| Power expended in the propulsion of the hull, . . . . | 606·13 | " 68·29 |
| Totals, . . . . . . | 887·60 | or 100·00 |

## THRUST OF THE SCREW.

The thrust of the screw during the foregoing maximum performance, calculated from the above data is as follows:

The power required to propel the simple hull is therein found to be 606·13 horses, equal to ($606{\cdot}13 \times 33000 =$) 20,002,290 pounds raised one foot high per minute. The speed of the vessel was 8·95 geographical miles of 6086 feet per hour or $\left(\frac{8{\cdot}95 \times 6086}{60} = \right)$ 907·82833 feet per minute; the resistance of the vessel at this speed, or its equivalent the thrust of the screw, was consequently $\left(\frac{20002290}{907{\cdot}82833} = \right)$ 22,033·119 pounds.

## PERFORMANCE AT SEA UNDER THE CONDITIONS OF ORDINARY PRACTICE.

There will be found in the three following Tables, Abstracts of the entire Steam Log of the "MINNESOTA" up to the present date, embracing the whole of her performance as recorded therein and during which all the particulars were noted. The performance has been divided into two parts; namely: 1st. That which is contained in the first Table and was done under Steam Alone. 2d. That which is contained in the second Table and was done under Steam and the Fore and Aft sails. 3d. That which is contained in the third Table and was done under Steam and the Square Sails combined.

The quantities in the Tables are the means from the Log, and were obtained in precisely the same manner as described under the head of the "MERRIMACK." All the other observations under that head in relation to the "Performance at Sea under the conditions of Ordinary Practice," apply equally here.

The vessel's draught of water, when leaving port with all her weights full and 620 tons of coal in the bunkers, was 22 feet 4 inches forward and 22 feet 10 inches aft, mean 22 feet 7 inches. As the coal was consumed the vessel lightened by the head, and when there were only 300 tons remaining the draught was 20 feet 1 inch forward and 22 feet 9 inches aft; mean, 21 feet 5 inches. The average draught of water for the entire steaming was 20 feet 11 inches forward and 22 feet 9 inches aft; mean, 21 feet 10 inches, to which the corresponding displacement is 4,455·4 tons with an accompanying greatest immersed transverse section of 808·2 square feet.

The variation in the draught of water, due to the consumption of the coal, was, it will be perceived, in the right direction and favorable to the propelling efficiency of the screw by keeping it always at its full immersion, with a continual sharpening of the water lines forward.

The performance recorded in the Tables, is that which was done during a two years cruise on the East India Station. It comprises an out and return voyage between the United States and Japan for the extreme points, with the incidental steaming on the Chinese coast.

The vacuum in the condensers, owing to air leaks, was very poor as an average for both engines. In one engine it was of ordinary goodness, that is about 26 inches of mercury, but greatly less in the other from the air leak through a large crack in the cylinder cover. With both engines in good order there was no difficulty in maintaining a vacuum of 26 inches of mercury.

The coal consumed during the cruise was nearly all Pennsylvania hard Anthracite sent out by the U. S. Navy Department.

The average temperature of the hot-well was 110° Fahr.; this, however, was not the temperature of the feed-water which, though taken from the hot-well, was pumped through the pipes of the heaters before being delivered into the boilers; and as these pipes were surrounded by the water *continuously* blown from the boilers through a small surface blow-cock to prevent the formation of scale, the feed-water received from this blown-out water sufficient heat to raise its temperature to an average of 135° Fahr., at which it entered the boilers. The extremes of this temperature were 125° and 145° Fahr. The heaters were situated by the side of the keelson and beneath the fire-room floor.

The density of the boiler water was maintained at 1¾ time the density of sea water. The heating surfaces of the boilers being everywhere easily accessible to cleaning tools, were kept perfectly free from scale and at their maximum evaporative efficiency throughout the cruise.

The three tables above referred to contain the performance of the vessel in detail under the conditions of wind, sea, and sail stated; the means of the quantities in these tables will be found in the table which follows them under the heading of "Synopsis of the Steam Log of the U. S. S. Frigate "MINNESOTA," together with the calculated results therefrom, made in the same manner as for the U. S. Screw Frigate "MERRIMACK."

## ABSTRACT OF THE STEAM LOG OF THE U. S SCREW FRIGATE "MINNESOTA," EMBRACING ALL HER PERFORMANCE UNDER STEAM ALONE.

| DATE | Number of consecutive hours. | Course of the Vessel. | WIND. Direction. | WIND. Kind. | STATE OF THE SEA. | Speed of the Vessel per hour, in geographical miles of 6086 feet. | Number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute. | Steam pressure in Boilers, in pounds per sq. inch above the atmosphere. | Vacuum in Condensers, in inches of mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Anthracite consumed per hour. | Per centum of the Coal lost, in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1857. | | | | | | | | | | | | | |
| May 30 & 31, | 27 | . . | Ahead. | Mod breeze. | Gentle swell. | 5·944 | 37·80 | 14·1 | 24·0 | 0·25 | 30·65 | 2959 | 20 |
| July 1 & 2, | 18 | . . | " | Fresh " | Mod. " | 5·192 | 36·00 | 15·0 | 21·0 | 0·20 | 36·40 | 2820 | 17¼ |
| July 2 & 3, | 15 | . . | On bow. | Light " | Gentle " | 6·533 | 38·52 | 16·0 | 24·8 | 0·20 | 25·20 | 2898 | " |
| " 11, | 24 | . . | " | Gentle " | " " | 6·042 | 36·15 | 14·0 | 25·5 | 0·16 | 26·29 | 2928 | 20 |
| " 12, | 8 | E. S. E. | S. E. | " " | " " | 6·125 | 36·00 | 12·0 | 25·0 | 0·16 | 24·97 | 3034 | " |
| " 29 & 30, | 29 | S. W. | S. | " " | Mod. " | 5·690 | 37·05 | 13·1 | 22·9 | 0·20 | 32·83 | 3177 | " |
| Aug. 3. 4. & 5, | 58 | S. E. by E | E. S. E. | Mod. " | Rough. | 4·310 | 36 44 | 18·0 | 22·0 | 0·25 | 47·84 | 3307 | " |
| " 17 & 18, | 22 | S. by E. | Abeam. | Light airs. | Smooth. | 7·682 | 40·86 | 15·0 | 28·0 | 0·22 | 17·09 | 3427 | " |
| Sept. 21 & 22, | 12 | S. W. | S. W. | " " | " | 7·750 | 41·50 | 15·0 | 28·5 | 0·22 | 17·64 | 3505 | 18¾ |
| Oct 23 & 24, | 17 | . . | . . | Calm. | " | 7·706 | 40·54 | 14·2 | 24·0 | 0·17 | 16·17 | 3126 | 19½ |
| " 24 & 25, | 11 | N. E. | S. E. | Light breeze. | " | 5·186 | 27·18 | 12·5 | 23·0 | 0·10 | 16·67 | 2269 | " |
| " 28, | 10 | N | W. | " airs. | " | 7·900 | 41·10 | 15·0 | 28·8 | 0·16 | 15·24 | 3744 | " |
| " 28 & 29, | 26 | N. E. | N. | " " | " | 7·923 | 41·70 | 15·0 | 28·6 | 0·20 | 16·20 | 3585 | " |
| Nov. 1, | 9 | W. by N. | S. by E. | Strong gale. | Heavy abeam | 4·555 | 35·55 | 15·0 | 24·0 | 0·16 | 43·50 | 2672 | 22 |
| " 3, | 21 | N. by E. | N. E. by N. | Mod. breeze. | Mod. swell. | 5·620 | 36·80 | 13·0 | 24·0 | 0·20 | 32·65 | 3275 | " |
| 1858. | | | | | | | | | | | | | |
| Feb. 27, 28, Mar. 1, 2 | 80 | S. E. | E. S. E. | Very light " | Smooth. | 7·800 | 41·95 | 12·6 | 24·0 | 0·19 | 18·00 | 3605 | 14 |
| March 14, | 24 | N. | N. | Gentle " | Mod. swell. | 6·844 | 40·22 | 14·4 | 23·0 | 0·20 | 24·95 | 4270 | 16⅘ |
| " 15, | 16 | N. by E. | E. N. E. | Strong " | Rough. | 4·469 | 36·00 | 15·0 | 23·5 | 0·23 | 45·25 | 3955 | " |
| " 16 to 22, | 102 | N. N. E. | N. E. | " " | " | 4·114 | 36·59 | 14·5 | 23·4 | 0·23 | 50·41 | 4067 | " |
| " 23, | 19 | N. | N. E. | Gentle " | Mod. swell. | 7·632 | 44·96 | 15·0 | 24·5 | 0·23 | 25·14 | 4266 | " |
| April 23, | 12 | N. | N. W. | Strong " | Heavy " | 8·605 | 35·33 | 15·0 | 24·0 | 0·20 | 55·00 | 3348 | 18¼ |
| " 24, | 24 | N. N. W. | N. N. W. | Light " | Rough. | 5·448 | 39·00 | 15·0 | 24·0 | 0·22 | 38·89 | 3482 | " |
| July 6 & 7. | 36 | E S. E. | E. S. E. | Gentle " | Smooth. | 6·775 | 40·26 | 11·2 | 24·0 | 0·22 | 25·79 | 3535 | 18⅗ |
| " 8, 9, & 10, | 64 | S. by W. | S. S. W. | Mod. " | " | 6·570 | 40·26 | 11·9 | 24·5 | 0·22 | 28·08 | 3581 | " |
| Aug. 5 & 6, | 34 | E. | E. by. S. | " " | Mod. swell. | 5·824 | 37·86 | 13·2 | 24·3 | 0·20 | 37·15 | 3233 | 18¾ |
| " 7, | 12 | E. by N. | N. E. | Strong " | Heavy " | 8·146 | 33·22 | 14·5 | 23·0 | 0·20 | 58·24 | 3420 | " |
| " 11, | 8 | E. | S S. E. | Light airs. | Smooth. | 7·562 | 40·54 | 13·0 | 23·0 | 0·17 | 17·73 | 3335 | 19 |
| Sept. 16, 17, & 18, | 60 | E. N. E. | E. N. E. | Mod. breeze. | Mod. swell. | 5·146 | 35·70 | 12·9 | 25·4 | 0·15 | 36·48 | 3018 | 22 |
| Dec. 20, 21, & 22, | 49 | N. W. | N. N. W. | Light " | Smooth. | 6·954 | 36·38 | 10·8 | 23·1 | 0·13 | 15·71 | 2465 | 18 |
| 1859. | | | | | | | | | | | | | |
| Jan. 8, | 6 | N. by W. | N. E. | " " | " | 5·667 | 31·40 | 10·0 | 23·5 | 0·11 | 20·41 | 2000 | 15 |
| " 12, | 3 | N. N. W | N. N. W. | " " | " | 6·800 | 33·97 | 10·0 | 24·3 | 0·11 | 18·21 | 2800 | 14 |
| " 13, 14, & 15, | 31 | N. N. W. | N. | " " | " | 6·755 | 35·75 | 10·0 | 24·8 | 0·11 | 16·67 | 2390 | " |
| " 15 & 16, | 33 | N. N. W. | N. N. W. | Gentle " | Gentle swell | 5·864 | 34·76 | 10·3 | 25·0 | 0·11 | 25·60 | 2240 | " |
| " 17, | 8 | N. | N. by E. | Light " | Smooth. | 6·000 | 31·96 | 9·9 | 24·8 | 0·09 | 17·21 | 2000 | " |
| Feb. 14, | 7 | W. N. W | N. N. W. | " " | " | 6·855 | 35·86 | 11·4 | 25·0 | 0·11 | 15·70 | 2300 | 12 |
| " 18, | 16 | W. N. W | N. W. by W. | " " | " | 6·891 | 36·50 | 10·1 | 25 0 | 0·11 | 16·73 | 2862 | " |
| March 16, | 4 | S. W. by S. | S. W. by W. | Gentle " | " | 5 000 | 38·75 | 10·0 | 25·5 | 0 11 | 34·66 | 2585 | 20½ |
| " 17 to 21, | 102 | S. W. | S. S. W. | Light " | " | 6·007 | 34·05 | 10·4 | 24·4 | 0·11 | 22·20 | 2691 | " |
| " 23 & 24, | 33 | S. W. | S. S. E. | Strong " | Rough. | 4·947 | 34·12 | 11·5 | 24·8 | 0·12 | 36·06 | 2844 | " |
| April 8, | 8 | Variable. | Variable. | Light airs. | Smooth. | 5·844 | 30·95 | 9·1 | 22·0 | 0·12 | 16·72 | 1900 | 20⅗ |
| May 12, 13, & 14, | 60 | N. N. W. | N. N. E. | " breeze. | " | 6·496 | 35·66 | 11·0 | 22·4 | 0·12 | 19·66 | 2977 | " |
| " 28 & 29, | 9 | N. W. | W. N. W. | Gentle " | " | 6·611 | 36·30 | 11·7 | 23·0 | 0·14 | 19·68 | 2658 | " |
| Means, | | | 25° from Ahead. | Gentle breeze. | Gentle swell. | 5·916 | 37·22 | 12·7 | 23·9 | 0·18 | 29·90 | 3190 | 18⅓ |

ABSTRACT OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "MINNESOTA," EMBRACING ALL HER PERFORMANCE UNDER STEAM AND FORE AND AFT SAILS.

| DATE. | Number of consecutive hours. | Course of the Vessel. | WIND. Direction. | WIND. Kind. | STATE OF THE SEA. | Speed of the Vessel per hour, in geographical miles of 6086 feet. | Number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute. | Steam pressure in Boilers, in pounds per sq. inch above the atmosphere. | Vacuum in Condensers, in inches of mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Anthracite consumed per hour. | Per centum of the Coal lost, in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1857. | | | | | | | | | | | | | |
| July 25 & 26, | 20 | S. E. | S. W. | Light breeze. | Gentle swell. | 7·600 | 40·65 | 18·1 | 24·5 | 0·22 | 17·54 | 8225 | 16 |
| Oct. 12 & 18, | 40 | N. N. E. | E. by N. | " " | " " | 8·125 | 41·68 | 15·0 | 24·0 | 0·16 | 14·03 | 8227 | 18$\frac{3}{4}$ |
| " 30 & 31, | 40 | N. E. | N. N. W. | Fresh " | Rough. | 7·400 | 41·10 | 15·0 | 24·0 | 0·20 | 20·59 | 8439 | 19$\frac{1}{2}$ |
| Nov. 1 & 2, | 39 | N. E. | E. S. E. | Mod. " | Mod. swell. | 6·641 | 37·49 | 14·7 | 24·3 | 0·14 | 21·88 | 2868 | 22 |
| " 4 & 5, | 34 | N. N. W. | N. E. | " " | " " | 6·595 | 39·00 | 15·0 | 28·0 | 0·18 | 25·42 | 3094 | " |
| 1858. | | | | | | | | | | | | | |
| March 13, | 10 | N. W. | W. | Light " | Smooth. | 8·225 | 42·80 | 14·4 | 28·5 | 0·20 | 15·25 | 4854 | 16$\frac{2}{3}$ |
| " 16, | 20 | N. N. E. | E. N. E. | Mod. " | Mod. swell. | 7·562 | 40·60 | 14·0 | 22·5 | 0·20 | 17·86 | 4277 | " |
| Sept. 19, | 16 | E. | N. E. | Light " | " " | 7·812 | 38·73 | 14·0 | 25·0 | 0·11 | 11·04 | 2976 | 22 |
| 1859. | | | | | | | | | | | | | |
| Jan. 11, | 9 | W. N. W. | N. by W. | Strong " | Smooth. | 7·778 | 38·60 | 11·1 | 28·5 | 0·09 | 11·14 | 2100 | 14 |
| " 12, | 7 | N. W. | W. | Light " | " | 7·357 | 38·00 | 11·3 | 24·5 | 0·11 | 14·62 | 2214 | " |
| " 13, | 12 | N. N. W. | W. | " " | " | 8·000 | 36·30 | 10·6 | 24·3 | 0·11 | 2·81 | 2125 | " |
| " 15, | 8 | N. N. W. | W. N. W. | Gentle " | " | 7·938 | 38·48 | 11·4 | 25·0 | 0·18 | 9·02 | 2564 | " |
| Feb. 14, 15, 16, 17, | 56 | W. N. W. | N. N. W. | " " | " | 7·216 | 35·74 | 11·0 | 24·9 | 0·10 | 10·96 | 2680 | 12 |
| " 19 & 20, | 14 | . . | On bow. | Light " | " | 7·286 | 36·09 | 11·0 | 25·2 | 0·10 | 10·96 | 2657 | " |
| March 15, | 8 | S. W. by S. | W. | Gentle " | " | 7·500 | 35·88 | 10·0 | 25·5 | 0·11 | 7·82 | 2596 | 20$\frac{1}{2}$ |
| " 16 & 17, | 11 | S. W. by W. | W. N. W. | Light " | " | 7·045 | 32·84 | 9·4 | 24·5 | 0·06 | 3·92 | 2876 | " |
| " 22, | 24 | S. W. | S. | Fresh " | Rough. | 4·888 | 33·90 | 11·2 | 24·8 | 0·11 | 37·18 | 2995 | " |
| April 21, | 7 | N. W. | W. | Light " | Smooth. | 7·071 | 36·26 | 12·0 | 24·5 | 0·10 | 14·00 | 2600 | 20$\frac{1}{3}$ |
| Means, | | | 57° from ahead. | Gentle breeze. | Gentle swell. | 7·213 | 38·25 | 13·1 | 24·2 | 0·14 | 16·83 | 3012 | 17$\frac{1}{3}$ |

ABSTRACT OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "MINNESOTA," EMBRACING ALL HER PERFORMANCE UNDER STEAM AND SQUARE SAILS COMBINED.

| DATE. | Number of consecutive hours. | Course of the Vessel. | WIND. | | STATE OF THE SEA. | Speed of the Vessel per hour, in geographical miles of 6086 feet. | Number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute. | Steam pressure in Boilers, in pounds per sq. inch above the atmosphere. | Vacuum in Condensers, in inches of mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Anthracite consumed per hour. | Per centum of the Coal lost, in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Direction. | Kind. | | | | | | | | | |
| 1857. | | | | | | | | | | | | | |
| July 3, | 6 | E. S. E. | Sd. & Ed. | Light breeze. | Gentle swell. | 6·666 | 36·50 | 13·0 | 25·0 | 0·20 | 19·45 | 2856 | 17¼ |
| " 12, | 11 | E. S. E. | S. W. by W. | Fresh " | Mod. " | 9·182 | 40·82 | 13·0 | 25·0 | 0·16 | 0·80 | 2523 | 20 |
| " 26, | 3 | S. E. by E. | S. by W. | Mod. " | " " | 8·920 | 44·33 | 12·0 | 22·0 | 0·14 | 12·25 | 2833 | " |
| " 27 & 28, | 9 | E. S. E. | S. | Fresh " | " " | 8·555 | 36·30 | 13·0 | 19·0 | 0·17 | – 3·81 | 2278 | " |
| " 30 & 31, | 29 | S. W. | S. S. E. | " " | " " | 9·347 | 39·90 | 14·0 | 22·5 | 0·17 | – 3·21 | 3142 | " |
| Sept. 22 & 23, | 21 | S. by W. | N. | Gentle " | Light " | 8·048 | 39·90 | 15·0 | 23·0 | 0·16 | 11·04 | 2867 | 18¾ |
| Oct. 11, | 7 | N. by E. | E. by N. | Light " | " " | 9·000 | 48·00 | 15·0 | 23·0 | 0·12 | 7·69 | 2441 | " |
| " 13 & 14, | 16 | N. by E. | E. | Gentle " | Smooth. | 9·250 | 39·88 | 14·8 | 23·8 | 0·16 | – 2·24 | 3185 | " |
| " 25, 26, & 27, | 67 | N. | S. W. | Light " | " | 8·254 | 40·60 | 14·7 | 23·8 | 0·16 | 10·34 | 3270 | 19½ |
| " 28, | 6 | N. E. | N. N. W. | Mod. " | " | 8·500 | 42·50 | 15·0 | 24·3 | 0·12 | 11·69 | 3315 | " |
| 1858. | | | | | | | | | | | | | |
| March 15, | 8 | N. by E. | E. by N. | Strong " | Rough. | 9·125 | 40·62 | 12·8 | 21·0 | 0·16 | 0·93 | 3692 | 16⅗ |
| " 17, | 3 | S. W. | Abeam. | Mod. " | Mod. swell. | 9·250 | 34·67 | 9·6 | 25·0 | 0·09 | –16·10 | 2471 | " |
| April 22, | 24 | N. | S. | " " | Smooth. | 9·885 | 41·38 | 15·0 | 23·0 | 0·13 | – 5·08 | 2922 | 18¼ |
| Oct. 7, | 14 | W. by S. | N. N. E. | " " | Rough. | 11·808 | 41·55 | 12·1 | 24·5 | 0·11 | –16·65 | 2544 | 16⅗ |
| " 8 & 9, | 28 | W. by S. | N. N. E. | Light " | Mod. swell. | 8·200 | 35·99 | 11·9 | 24·4 | 0·11 | – 0·48 | 2628 | " |
| Nov. 11 & 12, | 39 | . . | Abeam. | " " | Smooth. | 8·244 | 38·14 | 12·1 | 24·4 | 0·13 | 4·67 | 2157 | 17 |
| " 13 & 14, | 35 | S. W. ½ S. | N. E. | Strong " | " | 9·950 | 38·90 | 11·2 | 23·3 | 0·11 | –11·35 | 2067 | " |
| " 15, | 11 | W. | N. | " " | " | 9·409 | 38·51 | 11·0 | 24·5 | 0·13 | – 7 20 | 2000 | " |
| Dec. 8 & 9, | 12 | S. W. by S. | E. N. E. | Gentle " | " | 8·958 | 38·89 | 13·1 | 23·8 | 0·11 | – 1·56 | 2400 | 16 |
| " 14, 15, & 16, | 41 | S. S. W. | N. E. | Light " | " | 8·171 | 38·29 | 12·5 | 24·7 | 0·10 | 5·89 | 2070 | " |
| " 29, 30, & 31, | 47 | W. by N. | N. | " " | " | 7·819 | 36·83 | 10·9 | 23·6 | 0·11 | 6·37 | 2436 | 15 |
| 1859. | | | | | | | | | | | | | |
| Jan. 3 & 4, | 26 | . . | Abeam. | " " | " | 8·178 | 37·34 | 11·1 | 24·4 | 0·10 | 3·47 | 2428 | 17 |
| " 12, | 12 | W. N. W. | N. | Strong " | " | 8·500 | 40·26 | 11·5 | 24·5 | 0·11 | 6·90 | 2325 | 14 |
| Feb. 18 & 19, | 12 | W. N. W. | S. W. | Mod. " | " | 9·938 | 37·98 | 10·2 | 22·7 | 0·09 | –13·40 | 2467 | 16½ |
| " 21, | 10 | S. E. | N. N. W. | Gentle " | " | 8·900 | 36·76 | 10·0 | 25·5 | 0·09 | – 6·35 | 2090 | " |
| April 22, | 10 | N. W. | S. W. | Mod. " | " | 9·700 | 39·57 | 10·3 | 24·0 | 0·09 | – 7 50 | 2119 | 20⅗ |
| May 10 & 11, | 29 | N. W. | E. N. E. | Light " | " | 7·681 | 36·16 | 11·4 | 23·8 | 0·11 | 6·32 | 2372 | " |
| " 27 & 28, | 25 | N. N. W. | S. W. | Mod. " | " | 9·820 | 39·47 | 11·7 | 23·0 | 0·11 | – 8·86 | 2300 | " |
| | | | 116° from Ahead. | Gentle breeze. | Smooth. | 8·729 | 38·85 | 12·5 | 23·7 | 0·13 | 0·91 | 2574 | 17⅕ |

## SYNOPSIS OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "MINNESOTA."

| | | Performance under Steam alone. | Performance under Steam and the Fore and Aft Sails. | Performance under Steam and the Sq. Sails combined. | Mean of the three preceding columns. |
|---|---|---|---|---|---|
| | Total number of hours, | 1,162· | 375· | 561· | 2,098· |
| | Kind of wind, | Gentle breeze. | Gentle breeze. | Gentle breeze. | Gentle breeze. |
| | Angle made from ahead by the wind with the line of the vessel's keel, | 25° | 57° | 116° | 55° |
| | State of the sea, | Gentle swell. | Gentle swell. | Smooth. | . . |
| | Vessel's draught of Water, in feet and inches, Forward, | 20·11 | 20·11 | 20·11 | 20·11 |
| | Vessel's draught of Water, in feet and inches, Mean, | 21·10 | 21·10 | 21·10 | 21·10 |
| | Vessel's draught of Water, in feet and inches, Aft, | 22·9 | 22·9 | 22·9 | 22·9 |
| | Vessel's greatest immersed Transverse Section, in square feet, | 808·2 | 808·2 | 808·2 | 808·2 |
| | Vessel's Displacement, in tons, | 4,455·4 | 4,455·4 | 4,455·4 | 4,455·4 |
| | Mean Speed of the vessel per hour, in geographical miles of 6086 feet, | 5·916 | 7·213 | 8·729 | 6·900 |
| ENGINES. | Mean number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute, per counter, | 87·22 | 88·25 | 88·85 | 87·84 |
| | Mean steam pressure in the Boilers, in pounds per square inch above the atmosphere, | 12·7 | 13·1 | 12·5 | 12·7 |
| | Mean gross effective pressure on Pistons, in pounds per square inch, per indicator, | 9·75 | 9·00 | 7·27 | 8·95 |
| | Mean total pressure on Pistons, in pounds per square inch, per indicator, | 14·25 | 13·80 | 11·77 | 13·45 |
| | Gross effective horses power developed by the Engines, per indicator, | 542·18 | 514·82 | 421·98 | 505·98 |
| | Total horses power developed by the Engines, per indicator, | . . | . . | . . | 760·38 |
| | Vacuum in Condensers, in inches of mercury per gauge, | 23·9 | 24·2 | 23·7 | 23·9 |
| | Steam cut off at in Cylinders from commencement of stroke of piston, | ⅛ | ⅛ | ⅛ | ⅛ |
| | Proportion of Throttle-valve open, | 0·18 | 0·14 | 0·13 | 0·16 |
| | Slip of the Screw, in per centum of its speed, | 29·90 | 16·83 | 0·91 | 19·58 |
| | Temperature of the feed-water entering the Boilers, in degrees Fahr., | 185· | 185· | 185· | 185· |
| COAL. | Number of pounds of Coal consumed per hour, | 3,190· | 3,012· | 2,574· | 2,994· |
| | Per centum of waste of Coal in ashes, clinker, and fine coal, | 18⅛ | 17⅘ | 17⅘ | 18⅒ |
| | Number of pounds of Combustible consumed per hour, | 2,605·17 | 2,475·86 | 2,115·88 | 2,452·07 |
| | Number of pounds of Coal consumed per hour per square foot of grates, | 9·565 | 9·031 | 7·718 | 8 978 |
| | Number of pounds of Combustible consumed per hour per square foot of grates, | 7·811 | 7·424 | 6·844 | 7·358 |
| | Number of pounds of Coal consumed per hour per gross effective indicated horse power, | . . | . . | . . | 5·917 |
| | Number of pounds of Coal consumed per hour per total indicated horse power, | . . | . . | . . | 3·937 |
| | Number of pounds of Combustible consumed per hour per gross effective indicated horse power, | . . | . . | . . | 4·846 |
| | Number of pounds of Combustible consumed per hour per total indicated horse power, | . . | . . | . . | 3·225 |
| ANALYSIS OF ALL INDICATOR DIAGRAMS. | Mean steam pressure in Cylinders at commencement of stroke of piston, in pounds per sq. inch above zero, | 22·05 | 20·57 | 18·20 | 20·75 |
| | Mean steam pressure in Cylinders at the point of cutting off, in pounds per square inch above zero, | 18·43 | 17·20 | 15·22 | 17·35 |
| | Mean steam pressure in Cylinders at the end of the stroke of the piston, in pounds per square inch above zero, | 7·76 | 7·24 | 6·41 | 7·30 |
| | Mean back pressure in Cylinders against the pistons, in pounds per square inch above zero, | 4·5 | 4·8 | 4·5 | 4·50 |
| EVAPORATION FROM FEED-WATER TEMPERATURE 135° FAHR. | Pounds of Steam discharged per hour from the Cylinders, calculated from the pressure at end of stroke of piston, | . . | . . | . . | 15,888·280 |
| | Pounds of Steam condensed per hour in the Cylinders to produce the total power developed by the Engines, calculated from JOULE's equivalent, | . . | . . | . . | 1,972·685 |
| | Pounds of Steam that would have been evaporated per hour, had the heat been applied to evaporation that was lost by "blowing off," to maintain the sea water in the boilers at 1⅔ time the natural concentration, supposing the boilers to evaporate 9½ pounds of water per pound of anthracite, | . . | . . | . . | 3,484·267 |
| | Sum of the above three quantities, | . . | . . | . . | 21,345·182 |
| | Pounds of Water evaporated from temperature of Feed-water (135° Fahr.) per hour, supposing 9½ pounds of water vaporized by one pound of Anthracite, | . . | . . | . . | 28,443·000 |
| | Per centum of the Steam evaporated in the boilers not accounted for by the indicator, and by "blowing off," being the per centum which the difference between the quantities on the two preceding lines is of the quantity on the preceding line, | . . | . . | . . | 24·960 |

# "ROANOKE" AND "COLORADO."

## HULL.

In all other respects, except the following dimensions and proportions, the hulls of the "ROANOKE" and of the "COLORADO" are the same as that of the "MERRIMACK." The spars, sails, armament, &c. are precisely the same as those of that vessel.

| | |
|---|---|
| Length on mean load-water line from forward side of rabbet of stem to after side of rabbet of stern-post, | 263 feet 8½ inches. |
| Extreme breadth on mean load-water line, | 52 " 6 " |
| Moulded breadth on mean load water line, | 51 " 4 " |
| Depth from mean load-water line to lower edge of rabbet of keel, | 21 feet. |
| Depth of keel below lower edge of rabbet, | 2 " |
| Depth from top throat timbers to underside of spar-deck plank, amidships, | 33 feet 8 inches. |
| Depth from top throat timbers to underside of gun-deck plank, amidships, | 26 " 2 " |
| Depth from top throat timbers to underside of berth-deck plank, amidships, | 18 " 9 " |
| Displacement to mean load-water line, | 166,703· cubic feet. |
| Displacement to mean load-water line, | 4,772·22 tons. |
| Area of mean load-water line, | 11,268·3 square feet. |
| Displacement per inch of draught at mean load-water line, | 26 882 tons. |
| Area of greatest immersed transverse section to mean load-water line, | 902·9 square feet. |
| Centre of gravity of displacement before middle of the length of mean load-water line, | 3·03 feet. |
| Centre of gravity of displacement below load-water line, | 8·42 " |
| Angle of bow on mean load-water line, | 61° |
| Angle of stern on mean load-water line, | 73° |
| Angle of bow on water line at 9½ feet above lower edge of rabbet of keel, | 39° |
| Angle of stern on water line at 9½ feet above lower edge of rabbet of keel, | 31° |
| Angle of dead-rise of greatest immersed transverse section, | 12½° |
| Ratio of length to breadth on load water line, | 5·023 |
| Ratio of area of greatest immersed transverse section to area of circumscribing parallelogram, | 0·8190 |
| Ratio of area of load-water line to area of circumscribing parallelogram, | 0·8139 |
| Ratio of displacement to circumscribing parallelopipedon, | 0·5734 |

## MACHINERY.

The engines, boilers, and screw, are in all respects the precise duplicates of those of the "MINNESOTA," already described under that head.

## MAXIMUM PERFORMANCE IN SMOOTH WATER, UNINFLUENCED BY WIND OR CURRENT.

The following is the maximum performance that, uninfluenced by wind or current, can be permanently sustained in smooth water with the first quality of steam coal.

The different pressures in the cylinder is a collation of a large number of indicator diagrams.

| | |
|---|---|
| Vessel's mean draught of water, in feet, | 21·83 |
| Vessel's greatest immersed transverse section, in square feet, | 841·65 |
| Vessel's displacement, in tons, | 4,395·88 |
| Vessel's speed per hour, in geographical miles of 6086 feet, | 8·83 |
| Number of double strokes of engines' pistons, and of revolutions of the screw, made per minute, | 48·0 |
| Slip of the screw, in per centum of its speed, | 18·87 |
| Portion of the stroke of the piston from the commencement, at which the steam is cut-off, | 0·313 |
| Proportion of throttle-valve open, | Wide. |
| Steam pressure in the boilers, in pounds per square inch above the atmosphere, | 13· |
| Vacuum in the condenser, in inches of mercury, | 26· |
| Steam pressures in the cylinders, in pounds per square inch above zero, at the commencement of the stroke of the piston, | 25· |
| Steam pressure in the cylinders, in pounds per square inch above zero, at the point of cutting off, | 23· |
| Steam pressure in the cylinders, in pounds per square inch above zero, at the end of the stroke of the piston, | 8·2 |
| Mean back pressure in the cylinders, in pounds per square inch above zero, against the piston during its stroke, | 3·0 |
| Mean gross effective pressure on the piston during its stroke, in pounds per square inch, | 13·9 |
| Mean total pressure on the piston during its stroke, in pounds per square inch, | 16·9 |
| Gross effective horses power developed by the engines, | 996·826 |
| Total horses power developed by the engines, | 1211·968 |
| Temperature of the feed-water in degrees Fahrenheit, | 135· |
| Pounds of first quality steam coal consumed per hour, | 4250. |
| Per centum of refuse from the coal, in ashes, &c., | 12·5 |
| Pounds of coal consumed per hour per square foot of grate surface, | 12·561 |
| Pounds of combustible consumed per hour per square foot of grate surface, | 10·991 |
| Pounds of coal consumed per hour per gross effective indicated horse power, | 4·264 |
| Pounds of coal consumed per hour per total indicated horse power, | 3·506 |
| Pounds of combustible consumed per hour per gross effective indicated horse power, | 3·731 |
| Pounds of combustible consumed per hour per total indicated horse power, | 3·068 |

| | |
|---|---|
| Pounds of steam discharged per hour from cylinders, into condensers, calculated from the pressure of the steam at the end of the stroke of the piston, . | 22,505·638 |
| Pounds of steam per hour equivalent to the heat annihilated in the cylinders to produce the total power of the engines, calculated from Joule's equivalent, . | 3,154·085 |
| Pounds of steam that would have been evaporated per hour, had the heat been so applied which was expended in "blowing off" to maintain the sea water in the boilers at one and three-fourths time the natural concentration, supposing the boilers to evaporate 10 pounds of water per pound of coal, . . | 5,228·069 |
| Sum of the above three quantities, . . . . . . | 30,887·792 |
| Pounds of water evaporated from temperature of feed-water (135° Fahr.) per hour, supposing 10 pounds of water vaporized by one pound of coal, . | 42,500·000 |
| Per centum of the steam evaporated in the boilers not accounted for by the indicator and by "blowing off," being the per centum which the difference between the quantities on the two preceding lines is of the quantity on the preceding line, . . . . . . . . . | 27·32 |

## DISTRIBUTION OF THE POWER DURING THE ABOVE PERFORMANCE.

The pressure required to work the engines and shafting *per se*, being taken at 1½ pound per square inch of piston, the power thus absorbed is 107·57 horses.

Deducting from the gross effective power of 996·82 horses developed by the engines, this power of 107·57 horses, there remains 889·25 horses applied to the shaft, of which 7½ per centum or 66·69 horses is absorbed by the friction of the load.

The power expended in overcoming the cohesive resistance of the water by the screw blades, calculated in the ratio of the square of the velocity, and for a value of 0·45 pound avoirdupois per square foot of helicoidal surface moving in its helical path with a velocity of 10 feet per second, amounts to 78·04 horses.

The powers (66·69 and 78·04 horses) absorbed by the friction of the load and expended in overcoming the cohesive resistance of the water by the screw blades, being deducted from the power (889·25 horses) applied to the shaft, there remains 744·52 horses power expended in the slip of the screw and in the propulsion of the hull. And as the slip of the screw is 18·87 per centum of its speed, the power expended in it is (744·52×·1887=) 140·49 horses, leaving (744·52—140·49=) 604·03 horses expended in the propulsion of the simple hull.

Collecting the foregoing, we have the following distribution of the power; namely:—

| | Horses power. | Per centum. |
|---|---|---|
| Gross effective indicator power developed by the engines, . . . . | 996·82 | |
| Power required to work the engines and shafting *per se*, . . . . . | 107·57 | |
| Net power applied to the shaft, . . . . . . . . | 889·25 | or 100·00 |
| Power absorbed by the friction of the load, . . . . . . | 66·69 | 7·50 |
| Power expended in overcoming the cohesive resistance of the water by the screw blades, | 78·04 | 8·78 |
| Power expended in the slip of the screw, . . . . . . | 140·49 | 15·80 |
| Power expended in the propulsion of the hull, . . . . . | 604·03 | 67·92 |
| Totals, . . . . . . . . . . | 889·25 | or 100·00 |

## THRUST OF THE SCREW.

The thrust of the screw during the foregoing maximum performance calculated from the above data, is as follows:

The power required to propel the simple hull is therein found to be 604·03 horses, equal to $(604{\cdot}03 \times 33000 =)$ 19,932,990 pounds raised one foot high per minute. The speed of the vessel was 8·83 geographical miles of 6086 feet per hour or $\left(\frac{8{\cdot}83 \times 6086}{60} =\right)$ 895·65633 feet per minute; the resistance of the vessel at this speed, or its equivalent the thrust of the screw, was consequently $\left(\frac{19932990}{895{\cdot}65633} =\right)$ 22,255·177 pounds.

## PERFORMANCE AT SEA UNDER THE CONDITIONS OF ORDINARY PRACTICE.

In the two following tables will be found the entire performance of the "ROANOKE," as recorded in her steam logs at the Navy Department. The first table contains the performance under steam alone; the second table contains the performance under steam and the square sails combined.

The quantities in the tables are the means from the logs, and were obtained in precisely the same manner as described under the head of the "MERRIMACK." All the other observations under that head, in relation to the "Performance at Sea under the Conditions of Ordinary Practice," apply equally here.

For the steaming in the tables the vessel's mean draught of water was 21 feet 10 inches; the greatest immersed transverse section at that draught is 841·65 square feet, and the displacement 4,395·88 tons.

The coal used was almost entirely Pennsylvania anthracite; it was of average merchantable quality.

The poor vacuum in the condensers was due wholly to air-leaks. The average temperature of the hotwell was 110° Fahr.; the feed-water taken from it was raised to 135° Fahr., by being passed through the heaters before being forced into the boilers. The density of the boiler water was carried at 1¾ time that of sea water.

The two tables above referred to contain the performance of the vessel in detail under the conditions of wind, sea, and sail stated; the means of the quantities in these tables will be found in the table which follows them under the heading of "Synopsis of the Steam Log of the U. S. Screw Frigate 'ROANOKE,'" together with the calculated results therefrom, made in the same manner as for the U. S. Screw Frigate "MERRIMACK."

## ABSTRACT OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "ROANOKE," EMBRACING ALL HER PERFORMANCE UNDER STEAM ALONE.

| DATE. | Number of consecutive hours. | Course of the Vessel. | WIND. | | STATE OF THE SEA. | Speed of the Vessel per hour in geographical miles of 6086 feet. | Number of double strokes of engines' pistons, and of revolutions of the screw, made per minute. | Steam pressure in Boilers, in pounds per sq. inch above the atmosphere. | Vacuum in Condensers, in inches of Mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Anthracite consumed per hour. | Per centum of the Coal lost, in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Direction. | Kind. | | | | | | | | | |
| 1857. | | | | | | | | | | | | | |
| May 31, June 1, 2, 3, 4, | 44 | . . | On bow. | Gentle breeze. | Gentle. | 6·795 | 39·00 | 11·9 | 23·0 | 0·48 | 23·16 | 3360 | 17 |
| June 11 & 12, | 40 | S. | S. S. E. | " " | Moderate. | 5·900 | 34·65 | 15·5 | 22·0 | 0·38 | 24·91 | 2660 | " |
| July 21, 22, & 23, | 60 | N. N. W. | N. by E. | Light " | Gentle. | 6·700 | 38·75 | 12·8 | 23·0 | 0·50 | 23·75 | 3017 | " |
| " 26, | 24 | N. E. | E. N. E. | Gentle " | " | 7·625 | 40·58 | 13·0 | 21·5 | 0·44 | 17·14 | 3242 | " |
| " 27, | 24 | N. E. by E. | E. by N. | Fresh " | Rough. | 3·917 | 35·71 | 15·2 | 21·0 | 0·70 | 51·60 | 3117 | " |
| August 3, | 24 | N. | N. N. W. | Light airs. | Smooth. | 6·500 | 36·04 | 12·4 | 17·5 | 0·62 | 20·46 | 3038 | " |
| 1858. | | | | | | | | | | | | | |
| Sept. 19, | 18 | S. W. | S. W. | Gentle breeze. | Moderate. | 5·667 | 34·17 | 11·6 | 22·0 | 0·37 | 26·85 | 2244 | $20\frac{4}{5}$ |
| 1859. | | | | | | | | | | | | | |
| April 28, | 21 | E. by S. | N. E. | Light " | Smooth. | 5·333 | 35·00 | 12·0 | 20·0 | 0·44 | 32·80 | 2408 | $20\frac{1}{4}$ |
| May 26, | 12 | W. | S. W. | " " | " | 6·667 | 37·00 | 11·0 | 23·5 | 0·38 | 20·53 | 2750 | " |
| Oct. 4, 5, & 6, | 48 | N. W. | N. N. W. | " " | " | 4·938 | 31·44 | 12·0 | 18·0 | 0·30 | 30·78 | 2402 | $16\frac{4}{5}$ |
| " 9 & 10, | 30 | . . | On bow. | " " | " | 5·100 | 30·13 | 10·7 | 17·2 | 0·30 | 25·35 | 2208 | " |
| " 19, | 8 | W. by N. | . . | Calm. | " | 5·750 | 35·45 | 12·6 | 21·0 | 0·38 | 28·47 | 2275 | " |
| " 24, 25, & 26, | 30 | E. | E. | Light breeze. | " | 5·267 | 35·64 | 11·8 | 21·4 | 0·40 | 34·88 | 2433 | " |
| 1860. | | | | | | | | | | | | | |
| Feb. 8, 9, & 10, | 57 | N. | N. E. | Fresh " | Rough. | 4·105 | 35·36 | 12·4 | 20·0 | 0·62 | 48·80 | 3156 | $18\frac{1}{3}$ |
| " 15 & 16, | 48 | N. N. W. | N. | Gentle " | Smooth. | 7·000 | 41·88 | 11·8 | 20·0 | 0·63 | 25·40 | 3229 | $23\frac{4}{5}$ |
| April 27, 28, 29, & 30, | 64 | N. | N. N. E. | Mod. " | Rough. | 5·016 | 38·33 | 13·0 | 20·6 | 0·63 | 42·29 | 3300 | 20 |
| May 2, 3, 4, & 5, | 93 | N. | N. N. E. | Gentle " | Gentle. | 6·344 | 41·50 | 13·3 | 21·7 | 0·75 | 32·58 | 3525 | " |
| " 7, | 12 | N. E. | N. W. | Light airs. | Smooth. | 7·083 | 42·02 | 11·0 | 22·5 | 0·62 | 25·66 | 3025 | " |
| " 8, | 24 | N. by E. | N. N. E. | " breeze. | " | 6·750 | 42·00 | 13·0 | 19·0 | 0·62 | 29·13 | 3600 | " |
| 1861. | | | | | | | | | | | | | |
| July 1, 2, & 3, | 53 | S. S. W. | W. by S. | Mod. " | Moderate. | 5·236 | 35·17 | 6·9 | 23·5 | 0·50 | 34·34 | 2843 | 18 |
| " 10, | 9 | S. W. | S. W. | " " | " | 5·167 | 36·00 | 7·6 | 25·0 | 1·00 | 36·75 | 3522 | 21 |
| " 11 & 12, | 32 | . . | On bow. | " " | Rough. | 4·781 | 36·40 | 8·9 | 26·0 | 1·00 | 42·07 | 3394 | 15 |
| " 17 & 18, | 36 | S. S. W. | " | " " | " | 4·861 | 34·72 | 11·0 | 25·5 | 1·00 | 38·25 | 3250 | $13\frac{1}{2}$ |
| " 21 & 22, | 30 | N. N. W. | N. N. E. | Gentle " | Moderate. | 6·200 | 40·60 | 16·9 | 25·0 | 0·38 | 32·65 | 3600 | " |
| Means, | | | 32° from Ahead. | Gentle breeze. | Gentle. | 5·745 | 37·25 | 12·2 | 21·5 | 0·57 | 31·98 | 3053 | $18\frac{1}{10}$ |

## ABSTRACT OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "ROANOKE," EMBRACING ALL HER PERFORMANCE UNDER STEAM AND SQUARE SAILS COMBINED.

| DATE. | Number of consecutive hours. | Course of the Vessel. | WIND. Direction. | WIND. Force. | STATE OF THE SEA. | Speed of the Vessel per hour, in geographical miles of 6086 feet. | Number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute. | Steam pressure in Boilers, in pounds per sq. inch above the atmosphere. | Vacuum in Condensers, in inches of mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Anthracite consumed per hour. | Per centum of the Coal lost, in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1857. | | | | | | | | | | | | | |
| July 24 & 25, | 36 | N. N. W. | N. E. | Mod. breeze. | Smooth. | 8·722 | 41·17 | 12·9 | 22·9 | 0·44 | 6·57 | 3006 | 17 |
| July 31, Aug. 1 & 2, | 40 | N. N. E. | S. | Gentle " | " | 8·325 | 39·45 | 11·6 | 19·4 | 0·62 | 6·93 | 2888 | " |
| 1858. | | | | | | | | | | | | | |
| Sept. 12, 13, & 14, | 72 | S. by W. | E. | " " | " | 7·055 | 35·76 | 11·6 | 21·0 | 0·88 | 12·99 | 2361 | $22\frac{1}{5}$ |
| 1859. | | | | | | | | | | | | | |
| April 22 & 23, | 24 | W. by N. | N. | " " | Moderate. | 7·750 | 40·96 | 12·2 | 23·2 | 0·34 | 16·55 | 2608 | $20\frac{1}{4}$ |
| May 24 & 25, | 30 | W. S. W. | N. W. | " " | Smooth. | 6·500 | 37·23 | 12·0 | 23·2 | 0·38 | 23·01 | 2708 | " |
| October 17 & 18, | 33 | N. W. | S. | Light " | " | 5·394 | 33·85 | 12·0 | 18·6 | 0·38 | 29·73 | 2824 | $16\frac{4}{5}$ |
| 1860. | | | | | | | | | | | | | |
| Feb. 11, 12, 13, & 14, | 66 | N. N. W. | N. E. | Mod. " | Gentle. | 7·364 | 40·76 | 12·0 | 18·7 | 0·56 | 20·32 | 3870 | $18\frac{1}{5}$ |
| March 9 & 10, | 12 | S. by E. | N. N. W. | " " | Smooth. | 7·338 | 38·50 | 12·0 | 18·0 | 0·38 | 16·00 | 2500 | 24 |
| " 12 & 13, | 28 | S. S. E. | E. by N. | " " | Gentle. | 6·500 | 39·81 | 12·0 | 19·7 | 0·52 | 27·99 | 3111 | " |
| " 15, 16, & 17, | 52 | S. by E. | N. E. by E. | Gentle " | Smooth. | 8·040 | 41·20 | 9·9 | 18·2 | 0·62 | 13·94 | 3071 | " |
| April 30, May 1, | 34 | N. W. | N. E. | Mod. " | Moderate. | 8·382 | 44·42 | 14·7 | 20·0 | 0·63 | 16·78 | 3406 | 20 |
| May 7, | 12 | N. E. | S. | Light " | Smooth. | 8·500 | 43·80 | 12·0 | 21·0 | 0·63 | 14·41 | 3475 | " |
| " 9, | 14 | N. N. E. | E. | Fresh " | " | 10·714 | 49·31 | 13·5 | 20·0 | 0·63 | 4·18 | 3714 | " |
| 1861. | | | | | | | | | | | | | |
| July 18, | 8 | N. E. | N. W. | Mod. " | Moderate. | 7·375 | 35·62 | 16·7 | 25·0 | 0·25 | 8·70 | 2737 | $18\frac{1}{2}$ |
| " 19 & 20, | 20 | N. E. | S. | " " | " | 7·450 | 34·90 | 14·0 | 25·0 | 0·87 | 5·86 | 2645 | " |
| " 21 & 22, | 15 | S. E. | N. | " " | " | 7·000 | 39·13 | 12·0 | 25·0 | 0·87 | 21·10 | 2772 | " |
| Means, | | | 101° from Ahead. | Mod. breeze. | Gentle. | 7·536 | 39·43 | 12·2 | 20·6 | 0·49 | 15·71 | 2902 | $19\frac{7}{10}$ |

SYNOPSIS OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "ROANOKE."

| | Under Steam Alone. | Under Steam and Square Sails combined. | Mean of the two preceding Columns. |
|---|---|---|---|
| Total number of hours, | 844· | 496· | 1,340· |
| Kind of wind, | Gentle breeze. | Moderate breeze. | . . |
| Angle made from ahead by the wind with the line of the vessel's keel, | 82·° | 101·° | 57·5° |
| State of the sea, | Gentle. | Gentle. | Gentle. |
| Speed of the Vessel per hour, in geographical miles of 6086 feet, | 5·745 | 7·536 | 6·408 |
| Number of double strokes of Engine's Pistons, and of revolutions of the Screw, made per minute, | 37·25 | 39·43 | 38·06 |
| Slip of the Screw, in per centum of its axial speed, | 31·98 | 15·71 | 25·75 |
| Steam pressure in the boilers, in pounds per square inch above the atmosphere, | 12·2 | 12·2 | 12·2 |
| Vacuum in Condensers, in inches of mercury, | 21·5 | 20·6 | 21·2 |
| Proportion of Throttle-valve open, | 0·57 | 0·49 | 0·54 |
| Temperature in degrees Fahr. of the Hot-well, | 110·° | 110·° | 110·° |
| Temperature in degrees Fahr. of the Feed-water (after passing through the Heaters), | 135·° | 135·° | 135·° |
| Steam Pressure in Cylinders, per Indicator. In pounds per square inch above zero at commencement of Stroke of Piston, | 21·9 | 20·8 | 21·6 |
| In pounds per square inch above zero at point of cutting off the Steam, | 19·9 | 18·8 | 19·6 |
| In pounds per square inch above zero at end of Stroke of Piston, | 7·1 | 6·7 | 7·0 |
| In pounds per square inch above zero against the Piston during its stroke, | 5·3 | 5·8 | 5·5 |
| Mean Gross Effective Pressure in pounds per square inch on Piston during its stroke, | 9·9 | 8·5 | 9·4 |
| Mean Total Pressure in pounds per square inch on Piston during its stroke, | 15·2 | 14·3 | 14·9 |
| Gross Effective Horses Power developed by the Engines, | 550·96 | 500·736 | 534·515 |
| Total Horses Power developed by the Engines, | 845·93 | 842·415 | 847·263 |
| Pounds of Anthracite consumed per hour, | 3,053· | 2,902· | 2,997· |
| Pounds of Refuse from Anthracite per hour, in ashes, clinker, and dust, | 553· | 572· | 560· |
| Per centum of Refuse from the Anthracite, | 18·1 | 19·7 | 18·7 |
| Pounds of Combustible consumed per hour, | 2,500· | 2,330· | 2,437· |
| Pounds of Anthracite consumed per hour per square foot of grate surface, | 9·154 | 8·702 | 8·986 |
| Pounds of Combustible consumed per hour per square foot of grate surface, | 7·496 | 6 986 | 7·307 |
| Pounds of Anthracite consumed per hour per Gross Effective Horse Power, | 5·541 | 5·795 | 5·607 |
| Pounds of Anthracite consumed per hour per Total Horse Power, | 3·609 | 3·445 | 3·537 |
| Pounds of Combustible consumed per hour per Gross Effective Horse Power, | 4·537 | 4·653 | 4·559 |
| Pounds of Combustible consumed per hour per Total Horse Power, | 2·955 | 2·766 | 2·876 |
| Evaporated from temperature of Feed-water. Pounds of Steam discharged per hour from Cylinders into Condenser, calculated from the pressure of the steam at the end of the Stroke of the Piston, | . . | . . | 15,379·251 |
| Pounds of Steam per hour equivalent to the Heat annihilated in the Cylinders to produce the Total Power of the Engines, calculated from JOULE's equivalent, | . . | . . | 2,194·095 |
| Pounds of Steam that would have been evaporated per hour, had the Heat been so applied that was expended in "blowing off," to maintain the Sea-water in the Boilers at one and three-fourths time the natural concentration, supposing the Boilers to evaporate 9½ pounds of water per pound of Anthracite, | . . | . . | 3,464·697 |
| Sum of the above three quantities, | . . | . . | 21,038·043 |
| Pounds of Water evaporated from temperature of Feed-water (135° Fahr.) per hour, supposing 9½ pounds of water vaporized by one pound of Anthracite, | . . | . | 28,471·500 |
| Per centum of the Steam evaporated in the Boilers not accounted for by the Indicator, and by "blowing off," being the per centum which the difference between the quantities on the two preceding lines is of the quantity on the preceding line, | . . | . . | 26·18 |

## PERFORMANCE AT SEA OF THE U. S. SCREW FRIGATE "COLORADO," UNDER THE CONDITIONS OF ORDINARY PRACTICE.

In the following table will be found the entire performance of the "COLORADO" recorded in her logs during which the data is complete. It has been divided into two parts; the first comprising 414 hours done under steam alone; the last comprising 271 hours done under steam and the square sails combined. Beneath both will be found the means of the two.

All the remarks made in connexion with the "ROANOKE" apply in equal force to the "COLORADO," both as regards hull and machinery, which are in all respects precise duplicates of the "ROANOKE." The vessel's draught of water, and the manner of keeping the logs, &c., are the same as described for the "ROANOKE."

The cruising was done in the Gulf of Mexico and on the Atlantic coast of the United States. The fuel used was Pennsylvania anthracite.

It will be observed that, owing to air-leaks, the vacuum was very imperfect and the economic result low.

## ABSTRACT OF THE STEAM LOG OF THE U. S. SCREW FRIGATE "COLORADO," EMBRACING ALL THE PERFORMANCE RECORDED IN HER LOGS.

| DATE. | Number of consecutive hours. | Course of the Vessel. | WIND. Direction. | WIND. Kind. | STATE OF THE SEA. | Speed of the Vessel per hour in geographical miles of 6086 feet. | Number of double strokes of Engines' Pistons, and of revolutions of the screw, made per minute. | Steam pressure in Boilers, in pounds per sq. inch above the atmosphere. | Vacuum in Condensers, in inches of Mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Anthracite consumed per hour. | Per centum of the Coal lost in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **UNDER STEAM ALONE.** | | | | | | | | | | | | | |
| 1858. | | | | | | | | | | | | | |
| May 13 & 14, | 24 | S. E. by E. | E. N. E. | Light airs. | Smooth. | 6·000 | 36·49 | 12·8 | 22·8 | 0·25 | 27·48 | 3362 | 17 |
| " 17 & 18, | 36 | S. | On bow. | Light breeze. | " | 5·861 | 33 80 | 9·5 | 22·0 | 0·25 | 23·53 | 2978 | " |
| " 19, | 21 | S. by E. | S. S. E. | Gentle " | Moderate. | 5·143 | 33·13 | 9·5 | 20·5 | 0·25 | 31·54 | 2576 | " |
| " 21 & 22, | 48 | S. | S. S. E. | Moderate " | " | 5·750 | 37·05 | 11·2 | 24·0 | 0·25 | 31·56 | 2992 | " |
| 1861. | | | | | | | | | | | | | |
| June 23 & 24, | 30 | S. W. | S. W. | " " | " | 4·838 | 38·16 | 18·9 | 17·8 | 0·82 | 35·72 | 2479 | 18 |
| " 26 & 27, | 43 | S. W. | S. W. | " " | " | 4·744 | 32·37 | 11·1 | 18·1 | 0·44 | 35·37 | 2340 | " |
| July 1 & 2, | 15 | S. W. | S. | Gentle " | " | 5·067 | 33·73 | 12·2 | 15·5 | 0·30 | 33·76 | 1800 | 20½ |
| " 11, | 12 | W. | W. S. W. | " " | Smooth. | 4·917 | 37·33 | 13·0 | 19·0 | 0·50 | 41·91 | 2504 | 18½ |
| " 12 & 13, | 36 | N. W. | Abeam. | " " | " | 6·528 | 37·90 | 12·1 | 17·5 | 1·00 | 24·04 | 3162 | " |
| " 14, | 22 | N. W. | N. W. | Light " | " | 6·182 | 38·52 | 14·0 | 16·5 | 1·00 | 29·22 | 3003 | " |
| October 8, | 18 | N. N. E. | N. N. E. | Fresh " | " | 4·272 | 34·40 | 13·0 | 17·8 | 0·87 | 45·23 | 3167 | 16 |
| " 10 & 11, | 17 | E. | E. | " " | " | 5·300 | 36·59 | 13·5 | 17·5 | 0·70 | 36·11 | 3566 | " |
| November 7, | 10 | W. S. W. | S. W. | Light airs. | " | 5·700 | 38·86 | 12·1 | 19·0 | 0·87 | 34·46 | 3081 | 17 |
| 1862. | | | | | | | | | | | | | |
| June 10, 11, & 12, | 72 | S. E. | E. by N. | Fresh breeze. | Moderate. | 6·097 | 38·11 | 13·9 | 16·8 | 0·70 | 29·44 | 3021 | 16½ |
| " 21, | 10 | N. W. | W. | Moderate " | " | 5·800 | 39·70 | 14·0 | 16·5 | 1·00 | 35·55 | 2684 | " |
| Means, | | | 82° from Ahead. | Gentle breeze. | Gentle. | 5·593 | 35·95 | 12·3 | 19·0 | 0·55 | 31·89 | 2875 | 16½ |
| **UNDER STEAM AND SQUARE SAILS COMBINED.** | | | | | | | | | | | | | |
| 1858. | | | | | | | | | | | | | |
| May 28, | 12 | S. by W. | S. E. by E. | Mod. breeze. | Moderate. | 8·588 | 39·02 | 11·0 | 21·0 | 0·25 | 8·02 | 2167 | 17 |
| 1861. | | | | | | | | | | | | | |
| June 25, | 12 | S. S. W. | E. by S. | Gentle " | " | 7·000 | 36·87 | 11·8 | 19·0 | 0·37 | 16·27 | 2561 | 18 |
| July 2, 3, & 4, | 65 | W. by S. | S. S. E. | Mod. " | " | 6·877 | 32·80 | 9·5 | 15·0 | 0·87 | 7·53 | 1782 | 20½ |
| " 5, | 24 | S. W. | S. E. | " " | " | 8·542 | 36·90 | 10·0 | 15·5 | 0·37 | −2·09 | 2467 | " |
| " 6, | 8 | N. W. | E. | Fresh " | " | 8·250 | 39·38 | 10·2 | 15·5 | 0·50 | 7·60 | 2415 | " |
| " 12, | 12 | N. W. | S. W. | Mod. " | Smooth. | 7·167 | 37·33 | 12·0 | 18·5 | 0·81 | 15·32 | 2364 | 13½ |
| October 4, | 9 | S. W. | S. E. | Fresh " | " | 7·444 | 37·10 | 13·0 | 18·0 | 0·25 | 11·51 | 2458 | 16 |
| 1862. | | | | | | | | | | | | | |
| June 13, 14, & 15, | 65 | . . | Abaft beam. | Mod. " | " | 7·354 | 37·73 | 12·9 | 16·7 | 0·46 | 14·04 | 2871 | 16½ |
| " 17 & 18, | 24 | N. E. | S. S. E. | Fresh " | Rough. | 8·542 | 40·08 | 13·5 | 15 8 | 0·75 | 6·00 | 2716 | 21⅔ |
| " 19 & 20, | 40 | . . | On quarter. | Gentle " | " | 6·500 | 40·32 | 12·3 | 16·3 | 0·56 | 28·89 | 2355 | " |
| Means, | | | 107° from Ahead. | Mod. breeze. | Moderate. | 7·884 | 37·09 | 11·5 | 16·4 | 0·45 | 12·20 | 2378 | 19 |
| **MEANS OF THE WHOLE PERFORMANCE.** | | | | | | | | | | | | | |
| | . | . . | 62° from Ahead. | . . | . . | 6·302 | 36·40 | 12·0 | 18·0 | 0·51 | 23·65 | 2678 | 17½ |

# RELATIVE RESISTANCE OF THE SCREW

OF THE

## "MINNESOTA," "ROANOKE," AND "COLORADO,"

FOR EQUAL ROTARY SPEEDS, WITH THE VESSEL SECURED TO THE WHARF, AND WITH IT IN FREE MOTION IN SMOOTH WATER, UNINFLUENCED BY WIND OR CURRENT.

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WHILE the "ROANOKE" was lying at the wharf of the New York Navy Yard, drawing sensibly 21 feet 10 inches mean, of water, and working her machinery preparatory to going to sea; a number of indicator diagrams were taken from both ends of both her cylinders, and the number of revolutions per minute made at exactly the time the diagrams were taken was noted. The mean gross effective pressure per square inch of pistons, according to these diagrams, was 9·57 pounds; the number of revolutions made by the screw per minute was 27·66.

The screw of the "MINNESOTA" (exactly the same in all respects as that of the "ROANOKE") when propelling the vessel at her mean draught of 21 feet 10 inches in smooth water uninfluenced by wind or current, made 48·3 revolutions per minute with a mean gross effective pressure on the pistons of 13·8 pounds per square inch. The pressure required to work the engines *per se* was 1·5 pound per square inch of piston, which deducted from the 13·8 pounds, leaves 12·3 pounds per square inch applied to the screw shaft. Now as with equal conditions but unequal rotary speeds, the pressure applied to the screw shaft is in the ratio of the square of the speeds, and as the square of 27·66 is to the square of 48·3 as 1·0000 to 3·0492, the pressure 12·3 pounds would, for 27·66 revolutions of the screw per minute in free motion in smooth water uninfluenced by wind or current, become $\left(\frac{12{\cdot}3}{3{\cdot}0492} = \right)$ 4·034 pounds per square inch of piston.

With the vessel tied to the wharf, we have seen that this pressure, for 27·66 revolutions per minute, was (9·57 — 1·50 =) 8·07 pounds, or exactly double. Hence the resistance of the screw to rotation, when applied to the "MINNESOTA," was in one case exactly double that in the other. In this connexion it must be recollected that the normal slip of the "MINNESOTA'S" screw was 18·26 per centum.

## EXPERIMENT TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER OF THE "ROANOKE."

The following experiment was made upon one of the four boilers of the "ROANOKE" to determine its evaporative efficiency with Blackheath anthracite. The vessel lay alongside the wharf at the New York Navy Yard fitting for sea, and the evaporation was performed under the atmospheric pressure.

The boiler was perfectly tight, and it was covered with a thick coating of No. 1 felt stitched to No. 1 canvass; the top of the boiler to the sides was covered, additionally, with sheet lead. Each furnace door was perforated with four holes of 2 inches diameter for the admission of air, and they were kept open throughout the experiment.

The boiler had a safety-valve on the top of the shell; this valve was removed and the water introduced at its orifice by means of a hose pipe passing through a wooden bonnet bolted to the valve chamber. Although the water thus acted as a jet to condense some steam, yet the accuracy of the experiment was not thereby vitiated, because whatever heat was lost by the steam was gained by the water, and the temperature of the water being taken before it entered the boiler, the economic results would not be affected by the rise of its temperature afterwards due to the heat absorbed from the steam.

The stop-valve of the boiler was placed upon the steam chimney, and was 12 inches in diameter. This valve was removed, and a jointed iron pipe of 15 inches diameter and 28 feet height, bolted to the valve-chest nozzle, was used for the principal escape of the steam, a very small portion of which found its way through the safety-valve escape-pipe which was of copper and consisted of two parts, the first horizontal, 6 inches in diameter and 4 feet in length; the second 8 inches in diameter and 22 feet high. The joints of the large iron escape-pipe were open, the parts being only loosely attached, and the water which trickled out from them doubtless embraced all that was produced by the condensation of steam within. If any ran back to the boiler it, of course, diminished the apparent evaporation by that amount, which, if anything, was certainly an insignificant fraction. The portion of the steam passing through the safety-valve pipe was only that which escaped condensation by the entering water, and was so small as to be insensible for the principl part of the time, consequently, the condensation in this pipe must have been, practically considered, nothing.

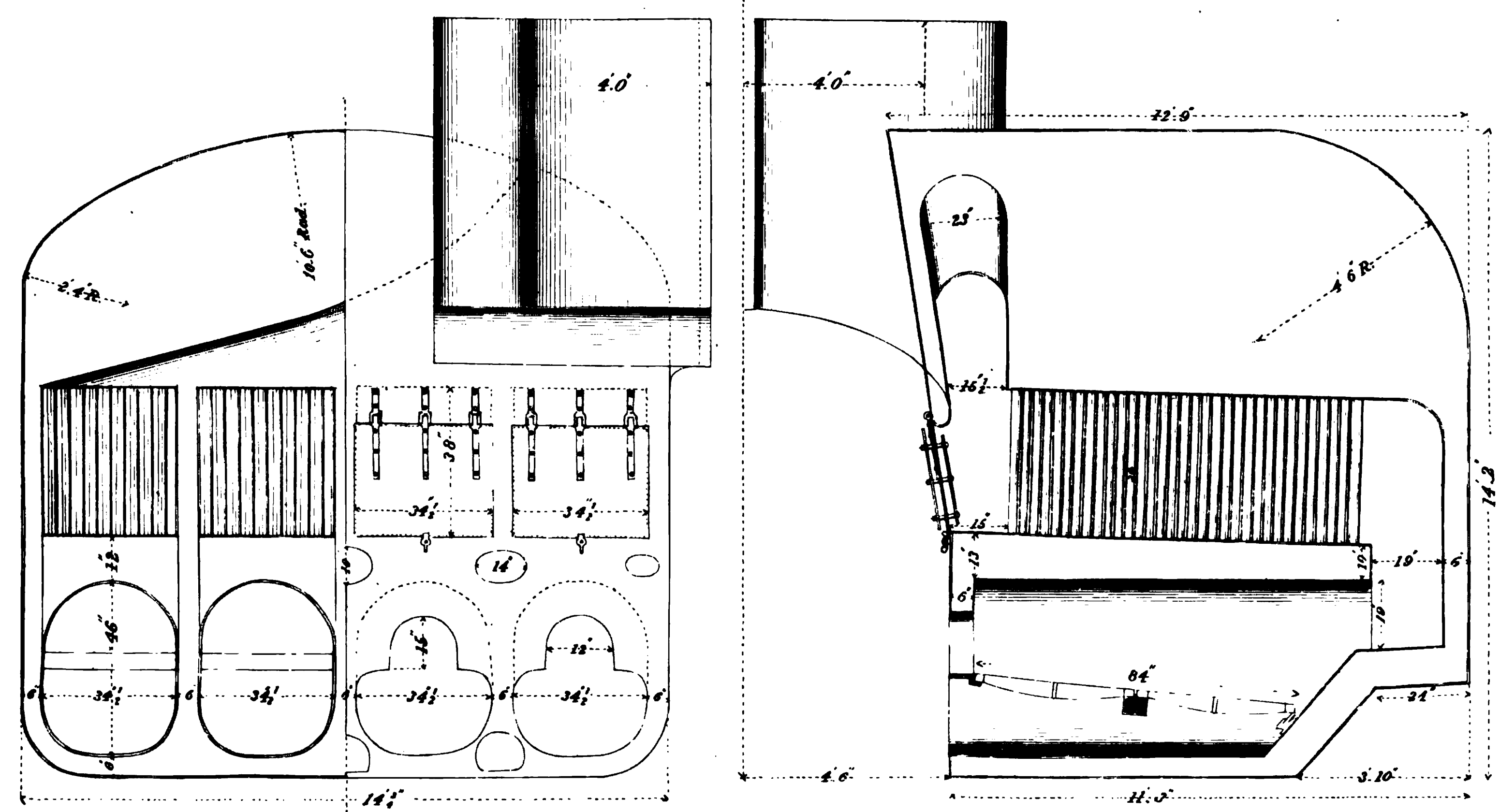
The water thus evaporated under the atmospheric pressure, was measured in a tank before being introduced into the boiler. This tank which was of wood lined with lead, was situated on the spar-deck of the vessel immediately over the boiler in use. It was filled at its top from the Navy Yard Reservoir through a hose having a pipe and stop-cock at its end; and was emptied from its bottom by gravity, through another hose passing to the chamber of the safety-valve as before described. This hose had a stop-cock bolted to the bottom of the tank. The tank, which was 8 feet long by 5 feet 6 inches wide, was filled each time to a mark which made the contents evacuated exactly 160 cubic feet.

The thermometer for ascertaining the temperature of the water remained continually submerged in the tank. The one for ascertaining the temperature of the products of combustion in the uptake was placed in a copper pot filled with oil and suspended over the centre furnace as near the tubes as possible without touching them, and about midway their height. It was taken out to be read, the oil preserving it from any sensible fall of temperature during the reading. The height of the water within the boiler was noted by a glass gauge.

Plate VII.

# Boiler of the U. S. Screw Frigates

## MERRIMAC, MINNESOTA, ROANOKE & COLORADO.




Half transverse section through furnaces, tube box, and uptake.

Half front elevation.

Longitudinal Section through centre of shell, furnace, and tube box.

All the coal consumed was carefully weighed in an iron bucket holding exactly 75 pounds, to which weight it was each time filled. The ashes, &c., were weighed in the same bucket and in the *dry* state.

The coal was fired as uniformly as possible, the fires were carried 10 inches thick, and were cleaned at regular intervals in precisely the same manner. As much coal was fed to the furnaces as they could consume, making the rate of combustion the maximum that could be obtained under the conditions of the experiment.

The experiment was conducted by the Engineer Department of the vessel, and in the following manner, the Assistant Engineers and Firemen standing regular watches as in practice at sea. The fires were started with sufficient wood to ignite the coal, and when a light clean coal fire was obtained, its thickness and the height of water in the glass gauge were noted, the time taken, and the experiment held to commence. At the end of the experiment the fires were thoroughly cleaned and made of the same thickness as at the commencement; and the height of water in the glass gauge brought up to the level at starting. Neither the wood nor the coal consumed previous to the time of considering the experiment begun, was included in the weight of coal producing the evaporation. The experiment continued exactly 72 consecutive hours, during which a tabular record was kept in which was noted at the end of each hour the temperature of the atmosphere on deck in the shade; of the water in the tank; of the fire-room; and of the products of combustion in the boiler uptake; the height of the barometer; and the weight of coal consumed. There were also noted in an appropriate column, the precise time when each tankful of water was emptied. The weight of ashes was recorded as they were withdrawn. The weather was described in the column of remarks.

With the experiment conducted in the manner narrated, the following results were obtained, namely:—

| | |
|---|---|
| Date of commencing the experiment, | Noon. May 24th, 1861. |
| Date of ending the experiment, | Noon. May 27th, 1861. |
| State of the weather, | For 12 hours light breeze and raining.<br>For 60 hours light breeze and clear. |

## TOTAL QUANTITIES.

| | |
|---|---|
| Duration of the experiment, in consecutive hours, | 72· |
| Number of cubic feet of water evaporated, | 6,216· |
| Number of pounds of water evaporated, | 387,381· |
| Number of pounds of Blackheath anthracite consumed, | 42,357· |
| Number of pounds of refuse from the anthracite, in ashes, clinker, &c., | 7,790· |
| Number of pounds of combustible consumed, | 34,567· |
| Per centum of the anthracite in refuse, | 18·39 |

## MEAN QUANTITIES.

| | |
|---|---|
| Temperature in degrees Fahr. of the external atmosphere on deck in the shade, | 65· |
| Temperature in degrees Fahr. of the fire room, | 77· |
| Temperature in degrees Fahr. of the water in the tank, | 60· |
| Temperature in degrees Fahr. of the products of combustion in the boiler uptake, | 240· |
| Barometer, | 29·81 |
| Thickness of the bed of anthracite upon the grates, in inches, | 10· |
| Pounds of anthracite consumed per hour per square foot of grates, | 7·056 |
| Pounds of combustible consumed per hour per square foot of grates, | 5·758 |

## EVAPORATION.

| | |
|---|---|
| Total number of pounds of water evaporated from temperature of 100° Fahr., | 401,757·864 |
| Total number of pounds of water evaporated from temperature of 212° Fahr., | 448,348·614 |
| Pounds of water evaporated from temperature of 100° Fahr. by one pound of anthracite, | 9·485 |
| Pounds of water evaporated from temperature of 212° Fahr. by one pound of anthracite, | 10·585 |
| Pounds of water evaporated from temperature of 100° Fahr. by one pound of combustible, | 11·623 |
| Pounds of water evaporated from temperature of 212° Fahr. by one pound of combustible, | 12·970 |

The results obtained when evaporating under the atmospheric pressure will be somewhat modified both absolutely and economically when the evaporation is performed under the boiler steam pressure, and other conditions of actual practice.

In the first place, as the experiment was made on only one of four boilers, the smoke pipe had an area four times too large, which greatly diminished the draught and, proportionably, the rate of combustion of the coal.

In the second place, the averege steam pressure in the boiler in actual practice was 12·2 pounds per square inch above the atmosphere or, say, a total pressure of 27 pounds, the temperature of which is, say, 244° Fahr. The temperature of the steam in the boiler during the experiment was, say, 212° Fahr., and as the temperature of the products of combustion on entering the uptake depends in measure on the temperature of the steam within the boiler, the temperature of the smoke-pipe and, consequently, the force of the draught will be greater from this cause in actual practice than under the experimental conditions.

Again, in consequence of the greater rapidity of the combustion, due to the above causes, a less *pro rata* amount of the heat will be taken up by the water and the excess will go to still further accelerate the combustion. The final result is found to be that, owing to the foregoing causes combined, the maximum rate of combustion is raised from 7 pounds of anthracite per hour per square foot of grate surface to 12 pounds.

The economic result in actual practice will be diminished from the same causes, a less proportion of the heat generated in the furnaces being taken up by the water, owing to the increased temperature of the products of combustion in the smoke-pipe.

The temperature of these products under the conditions of actual practice, burning, say, 9 pounds of coal per hour per square foot of grate surface, is about 300° Fahr., or 60° Fahr., above that of the experiment. Assuming, now, the temperature of the furnaces to average 2000° Fahr., the reduction of economic efficiency would be 3 per centum due to increased temperature of the products of combustion in the smoke-pipe.

The heat imparted to the water above 100° Fahr. during the experiment was (1177·79 — 100 =) 1077·79° Fahr., and the evaporation from 100° Fahr. was 9·485 pounds of water per pound of coal. The average temperature of the feed-water in actual practice was 135° Fahr., and the heat imparted was (1188 — 135 =) 1053° Fahr., making a gain in economic efficiency of $\left(\frac{1077 \cdot 79 - 1053 \cdot 00 \times 100}{1077 \cdot 79} = \right)$ 2·3 per centum or, say, offsetting the above loss of 3 per centum, and leaving the evaporation from water of 135° Fahr., temperature under the conditions of actual practice, say, 9½ pounds of water per pound of coal.

The per centum of the coal in refuse during the experiment was about the average that obtains in actual practice.

## GENERAL REMARKS.

In comparing the economic results from the machinery of the Frigates, we find, referring to the distribution of the power, that the losses of useful effect by the screws of all of them except by that of the "MERRIMACK," when propelling at the maximum speed in smooth water uninfluenced by wind or current, averaged 24·025 per centum of the power applied to their shafts. This loss was composed of slip 15·42 per centum, and of power expended in overcoming the cohesive resistance of the water by the screw blades, 8·605 per centum. The corresponding losses by the screw of the "MERRIMACK" were slip 20·21 per centum, and power expended in overcoming the cohesive resistance of the water to the screw blades 4·40 per centum; total 24·61 per centum, or very nearly the same as the others. It thus appears that the less cohesive resistance of the water to the screw blades of the "MERRIMACK," due to their less helicoidal surface and less helical speed, almost exactly compensated in economic result their greater loss by slip, due to the same less helicoidal surface and less helical speed.

If we compare the slip of the screws of the "WABASH" and "MINNESOTA" at sea under the conditions of ordinary practice, we shall find it to average for the performance under steam alone 29·7 per centum of their speed; while in the case of the "MERRIMACK'S" screw this slip was 41·74 per centum. Now the ratio of the slips of the screws of the "WABASH" and "MINNESOTA" in smooth water uninfluenced by wind or current, and at sea under the conditions of ordinary practice and for the performance under steam alone, was as 1·0 to 1·6. The same ratio applied to the "MERRIMACK'S" screw would make it 36·7 per centum of its speed instead of the above 41·74; hence the adverse influence of the weather in the case of the "MERRIMACK" was one-seventh greater than in the case of the "WABASH" and "MINNESOTA." From these figures it will also appear that the screw of the "MERRIMACK" was a few per centum less economical than the screws of the "WABASH" and "MINNESOTA" at sea under the conditions of ordinary practice.

The average consumption of coal by the frigates at sea was about 3230 pounds per hour, or 34·6 tons per 24 hours; and their average speed with their machinery in tolerable order, about 7 knots per hour, with and without the assistance of sail. Under these circumstances they could steam 430 hours and make 3010 knots. The average indicated gross effective horses power developed was 580, and was obtained at an average cost of 5·57 pounds of coal per horse power per hour. This was a very large expenditure, especially when the high evaporative results from the boilers are considered, and was due to the large capacity of the cylinders in proportion to the weight of steam used through them—which caused a low average *total* pressure upon the piston during its stroke—and to the high back pressure due to the very imperfect vacuum averaged. To appreciate the largeness of the influence exerted on the economic result

by these causes, we have only to consider that, for the average performance at sea, the mean total pressure was 15·5 pounds per square inch of pistons; of this 5·5 pounds were back pressure and 1·5 pound was the pressure required to work the engines *per se*, leaving only $(15{\cdot}5 - \overline{5{\cdot}5 + 1{\cdot}5} =)$ 8·5 pounds per square inch or $\left(\frac{8{\cdot}5 \times 100}{15{\cdot}5} = \right)$ 54·84 per centum applied to the screw shaft. Let us now suppose the area of the cylinders to have been reduced one-half and the back pressure to have been $3\frac{1}{2}$ pounds per square inch, other things remaining the same, then the total average pressure would have risen to $(15{\cdot}5 \times 2 =)$ 31 pounds per square inch, and the sum of the back pressure and of the pressure required to work the engine would have been $(3{\cdot}5 + 1{\cdot}5 =)$ 5 pounds, leaving $(31 - 5 =)$ 26 pounds per square inch or $\left(\frac{26 \times 100}{31} = \right)$ 83·87 per centum applied to the screw shaft, instead of 54·84 per centum; an immense gain, and illustrating the real cause of the economy of using high pressure steam and a good vacuum.

The maximum performance that could be permanently maintained in smooth water uninfluenced by wind or current, and with the machinery in perfect order, was 9 knots per hour with a consumption of 4,250 pounds of first quality steam coal, or at the rate of 45·54 tons per 24 hours. The indicated gross effective power was 1000, and was obtained at the cost of 4·25 pounds of coal per hour per horse power. It will be observed that this performance includes the losses due to cleaning fires, and to "blowing off" sufficiently to maintain the sea water in the boilers at $1\frac{2}{3}$ time its natural concentration; this latter loss alone amounted, under the conditions, to $12\frac{1}{2}$ per centum of the weight of fuel consumed. Such a maximum performance is a very different thing from a "run at the measured mile," the results of which are really a *maximum maximorum*, instead of a maximum, and are only determinable of sufficiency of cylinder capacity, not of boiler power. At the above rate of speed and consumption of coal, the frigates could steam 326 hours and make 2,934 knots.

At the above maximum performance, the number of double strokes of piston made per minute was 48; and for the mean at sea 39. The latter number was quite as great as the engines could with safety bear. Neither the strength of the parts, nor the size of the journals, nor the openings of the various valves, pipes, and cocks, would admit of a greater speed for any length of time. The engines were much too light for direct-acting screw engines; they had only the proportions of paddle-wheel engines and gave much trouble when put to any speed. The crank-shafts broke, the main-journals and crank-pin journals were in a chronic state of heat and required the water to be started on them with the engines, the air-pump valve-seats were broken from the flanges, to which they were bolted; eccentric straps and other details also gave way; and the trunks in the trunk engines were continually breaking. The brass lining for the air-pumps was put in in staves, as with paddle wheel engines, and in every case the staves tore out, and presented so rough a surface that no packing could last but a few hours. In the one case the air-pump was horizontal, in the other it was vertical, which made all the difference between what would and what would not do. Only a solid brass casting will answer for a lining in a horizontal position. The main-bearings and crank-pin journals were deficient fully one-half in surface.

Of the different frigates' engines, the performance of those of the "WABASH" was the most satisfactory, and arose mainly from the fact of their being erected on a deep, strong bed-plate extending beneath them their whole length athwartship; this preserved all the parts of the same engine in rigid con-

nexion and true line with each other. The engine room in this vessel was between the engines, instead of being on a gallery over them, and left all the working parts easily accessible and always under inspection when in operation. In the other engines, the important working parts were out of sight, hid beneath the engine room floor, and were only visible to the oiler crawling among them with a feeble lamp. These engines had no bed-plate, nor sufficient stiffness in their frames and cylinders to compensate the want; the main pillow-blocks had no connexion other than the shaft, fore and aft the vessel, the condenser casting being separate for each engine and bolted independently to the vessel. With large engines thus formed of disconnected parts bolted separately to the vessel, great precaution is required in obtaining in the vessel itself, a rigid, unyielding platform of wood, composed of large timbers strongly bolted to the vessel's frames, lying close together, bolted to each other and caulked between, and extending under the entire area occupied by the engines, forming, in fact, one large wooden bed-plate for them. It is essential in quick working screw engines that all their parts be rigidly connected, and the strength for this must be obtained in extra dimensions of the detail, and in forming the principal parts into one unyielding mass, or in extra precaution in giving strength to that part of the vessel itself which is to sustain them; the former is the safest plan and the best, it is making the engines hold the vessel instead of the vessel holding them; and its benefits become most apparent as the vessel grows old and shaky.

In the trunk engines, the trunks gave great trouble, they were difficult to keep air-tight, and there resulted a ruinous loss of vacuum. Unless very carefully packed they heated, and then the application of water warped and cracked them. The connecting-rod pin within the trunk—a very important journal—was practically inaccessible.

The analyses of the indicator diagrams from the frigates' engines show the additional condensation in the cylinder due to the trunk. With the engines of the "MERRIMACK" and "WABASH," which are of the back-action kind, the per centum of the steam evaporated in the boilers, not accounted for by the indicator, is 20; while with the trunk engines it is 26; the difference, of course, being due the greater surface exposed to external refrigeration by the cylinder and its trunk, particularly by the latter, which being of metal only 1 inch thick and unprotected by any covering, and being alternately thrust out into the cool air and withdrawn back into the cylinder, and having its whole inner surface rapidly moving through the air, must act as a powerful refrigerator.

The great fault in the proportioning of the machinery consisted, as previously noticed, in the enormous cylinder capacity which caused the steam to have a low average pressure upon the piston throughout its stroke. Could steam engines work without friction and without back pressure against the piston, this low average pressure would be of but little, if any, consequence; but as both exist and are constant for the same engine, they become a larger proportion of the total average pressure as that becomes less. As the back pressure, in the case of the frigates' engines was very large, owing to air-leaks, the evil effects of the large capacity of cylinder were so greatly exaggerated that, notwithstanding the high evaporation given by the boilers, the economic results realized at sea under the conditions of ordinary practice were very low.

Another great fault consisted in the fact that the vessels were under-powered for good economic results when steaming at sea under the conditions of ordinary practice. It will be recollected that they were frigate built and full rigged ships; they were high out of water and exposed in their hulls, spars and rigging, a very great surface to air resistance. When steaming in smooth water and in a calm, or with the wind, the evil effects of the under power did not appear, but immediately the vessel was brought head to

wind, or encountered a rough head sea, its resistance increased enormously and, the slip of the screw rising in proportion, the speed fell to mere steerage way, while the consumption of fuel continued nearly the same.

No satisfactory result can be obtained from vessels of the frigate type, if powered for a less speed than 10 knots per hour in smooth water uninfluenced by wind or current; and the screw under these conditions, should be proportioned to have a slip not exceeding 10 per centum. In the case of the frigates, had the capacity of the cylinder been reduced about five-eighths and the steam allowed to follow the piston as far, at least, as half stroke; and had the pitch of the screw been expanding, with a mean of about 16 feet; a very much greater effect would have resulted, both economically and potentially; and at a less cost of machinery. With these changes, the maximum number of revolutions of the screw would have risen to about 70 per minute, and the maximum speed of the vessel in smooth water to about 9$\frac{3}{8}$ knots per hour; the boilers, their maximum pressure, and their consumption of fuel remaining the same.

The machinery of the "MERRIMACK" was built by the West Point Foundry, Cold Spring, New York. That of the "WABASH" by MERRICK & SONS of Philadelphia, Pennsylvania. That of the "MINNESOTA" was constructed at the Navy Yard, Washington, D. C. And that of the "ROANOKE" and "COLORADO" by the Tredegar Works, Richmond, Virginia. The average cost of the machinery for each vessel was $170,000.

UNITED STATES SCREW SLOOP

"BROOKLYN."

2 D

# UNITED STATES SCREW SLOOP "BROOKLYN."

THE "BROOKLYN" is a large screw sloop built for the United States Navy in 1858 at the private ship-yard of JACOB A. WESTERVELT, New York, and under the superintendence of Naval Constructor S. A. POOK. The machinery was constructed by JAMES MURPHY & Co., of the same city. The crew numbers about 300 men. Weight of coal carried in bunkers 360 tons.

The following are the dimensions of the hull and machinery, to which are added the maximum performance of the vessel under normal conditions, and abstracts of her steam logs embracing all the performance at sea therein recorded up to the present date.

## HULL.

The vessel has full but easy water lines, and well rounded bilges.

| | |
|---|---|
| Length on load-water line from forward side of rabbet of stem to after side of forward stern-post, | 233 feet. |
| Extreme breadth on load-water line, | 43 " |
| Moulded breadth on load-water line, | 42 " |
| Depth of hold, | $22\frac{2}{3}$ " |
| Depth from rabbet of keel to load-water line, | 15 " |
| Depth of keel below rabbet, | 1 foot 2 inches, |
| Deep load draught of water, | 16 feet 2 " |
| Area of greatest immersed transverse section, | 579·66 square feet. |
| Area of load-water line, | 8094· " " |
| Displacement, | 2686 tons. |
| Displacement per inch of draught at load-water line, | 19 309 tons. |
| Weight of hull, | 1350 tons. |
| Centre of gravity of displacement below load-water line, | 6·50 feet. |
| Metacentre above centre of gravity of displacement, | 10·44 " |
| Ratio of greatest immersed transverse section to circumscribing parallelogram, | 0·8987 |

| | |
|---|---|
| Ratio of load-water line to circumscribing parallelogram, | 0·8078 |
| Ratio of displacement to circumscribing parallelopipedon, | 0·6243 |
| Angle of entrance at load-line, | 56° |
| Angle of clearance at load-line, | 80° |
| Angle of entrance at 7½ feet above rabbet of keel, | 39° |
| Angle of clearance at 7½ feet above rabbet of keel, | 38° |
| Angle of dead-rise at greatest transverse section, | 3¼° |
| Surface of the plain sails, in square feet, | 22,480. |

## BATTERY.

The battery is composed of sixteen shell guns of 9 inches bore, carried in broadside; and of two shell guns of 10 inches bore, mounted on pivots, one forward and the other aft. There are ports enough, however, to carry twenty-four guns in broadside.

| | |
|---|---|
| Weight without carriages of the two 10-inch guns, | 23,890 pounds. |
| Weight without carriages of the sixteen 9-inch guns, | 146,916 " |
| Total weight of all objects of ordnance including the above guns, | 406,113 " |

## ENGINES.

Two horizontal, condensing, direct-action engines, with the connecting-rod extending directly on from the crosshead. The cylinders are placed on one side the keelson, and have their condensers and pumps on the opposite side. There are but three frames for both engines. Each cylinder has two piston-rods upon the same level secured into the crosshead, and the connecting-rod journal is between them. The air-pump and the feed-pump are worked direct from arms or lugs forged on the crosshead: the arm for the air-pump is 12 inches from centre of crosshead to centre of pump-rod, and that for the feed-pump is 10 inches between the same points. The air-pump is double-acting, the feed-pump single-acting. The steam-valve is a packed three-ported slide placed horizontally on the top of the cylinder and worked by a Stephenson link through a rock-shaft. The cut-off valves are two plates worked on the back of the steam-valve by a separate eccentric and rock-shaft. They are attached to the same valve-rod, and adjustible by right and left hand screws. The thrust-bearing consists of six collars of 15 inches exterior diameter, and one collar of 16 inches exterior diameter placed on a shaft of 12 inches diameter, and presenting an effective area of 470 square inches.

| | |
|---|---|
| Diameter of the cylinders, | 61 inches. |
| Stroke of the pistons, | 33 " |
| Space displacement of both pistons per stroke, exclusive of bulk of rods, | 110·874 cubic feet. |
| Diameter of piston-rod (two to each cylinder), | 5 inches. |
| Area of steam-port, (4½ by 36 inches), | 162 square inches. |
| Area of exhaust-port, (6½ by 36 inches), | 234 " |
| Area of steam-opening through steam-valve for cut-off ports (4 by 34 inches), | 136 " |
| Clearance, | $\frac{7}{16}$-inch. |
| Steam space at *one* end of *both* cylinders in nozzles and clearance, | 6·168 cubic feet. |
| Diameter of air-pump (double-acting), | 19 inches. |
| Stroke of air-pump piston, | 33 " |

| | |
|---|---|
| Area of air-pump receiving-valves, | 207 square inches. |
| Area of air-pump delivery-valves, | 234 " |
| Diameter of feed-pump (single-acting), | 5½ inches. |
| Stroke of feed-pump piston, | 33 " |
| Diameter of injection-valve, | 5 " |
| Diameter of outboard delivery-valve, | 17 " |
| Diameter of crank-shaft journals, | 13 " |
| Length of crank-shaft journals, | two of 22 and one of 26 " |
| Diameter of crank-pin journals, | 12 " |
| Length of crank-pin journals, | 12½ " |
| Diameter of screw-shaft journals, | 12 " |
| Length of screw-shaft journals, | 18 " |
| Diameter of crosshead journals, | 9 " |
| Length of crosshead journals, | 12½ " |
| Area of each guide-gib of crosshead (6 by 15 inches), | 90 square inches. |
| Diameter of connecting-rod in neck, | 6½ inches. |
| Total weight of engines, including spare pieces, | 191,500 pounds. |
| Length of engines over all, in fore and aft direction of vessel, | 16 feet. |
| Breadth of engines over all athwartship, | 24 " |
| Height of engines over all, | 10 " |

## BOILERS.

Two vertical water-tube boilers with the tubes placed above the furnaces. The boilers are opposite each other with a fire room 9 feet wide between them, and extending in the fore and aft direction of the vessel. Both boilers have one telescopic chimney in common placed at their centre and above the fire room. The tops of the furnaces and the bottoms of the ash-pits are semicircular. The tubes of each row lengthways the furnaces have their centres not in the same straight line, but in two straight lines, diverging from the centre of the first tube at the back end of the row ⅜ths of an inch at the centre of the last tube at the front or chimney end of the row: the alternate tubes have their centres in these two lines. The four corner tubes are omitted for convenience of construction. The tubes are of brass and seamless.

Each boiler is provided with a heater—a cast iron cylindrical vessel of 12½ inches outside diameter, containing thirty-one brass tubes of 1¼ inches external diameter and 12 feet length. The supersalted water of the boiler is blown off continuously from the surface by a cock and pipe, and passes around these tubes on its way to the sea; while the continuous feed passing through these tubes on its way from the hot-well to the boiler, receives from the blown out water an accession of about 30° Fahr. of temperature.

The boilers, including the fire room between them, occupy on the floor 23 feet 10 inches in the fore and aft direction of the vessel, and 31 feet in the athwartship direction. They have a steam chimney which surrounds the smoke chimney and is 2 feet high above the top of the shell. On this steam chimney the stop and safety-valves are placed.

| | |
|---|---|
| Length of each boiler (athwartship) at furnaces, | 11 feet. |
| Length of each boiler (fore and aft the vessel) at top, | 11 " 9 inches. |
| Breadth of each boiler (fore and aft the vessel), | 23 " 10 " |
| Height of each boiler (exclusive of steam chimney), | 11 " 3 " |

| | |
|---|---|
| Number of furnaces in each boiler, | 7. |
| Breadth of each furnace, | 2 feet 9 inches. |
| Length of fire grates, | 6 " 6 " |
| Total area of grate surface in both boilers, | 250¼ square feet. |
| Height from bottom of ash-pit to crown of furnace, | 42 inches. |
| Number of rows of tubes lengthways the furnace, | 30. |
| Number of rows of tubes crossways the furnaces, | 9. |
| Total number of tubes in both boilers, | 3724. |
| Length of space lengthways the furnace occupied by the tubes, | 7 feet 1 inch. |
| Depth of space between crown of furnace and the lower tube-plate at front, | 12 inches. |
| Depth of space between crown of furnace and the lower tube-plate at back, | 10 " |
| Space in the clear between the tubes, lengthways the furnace, | 0·862 inch. |
| Space between the tubes for direct draught at back end of furnaces, | 1·500 " |
| Space between the tubes for direct draught at front or chimney end of furnaces, | 1·125 " |
| Length of tubes, extreme, | 33 inches. |
| Length of tubes in clear between plates, | 32 " |
| External diameter of tubes, | 2 " |
| Calorimeter or cross area between the tubes for direct draught, at back end of both boilers, | 46⅔ square feet. |
| Calorimeter or cross area between the tubes for direct draught, at front end of both boilers, | 35 " |
| Total area of heating surface in both boilers, measuring the tubes on their exterior circumference, | 7788 " |
| Diameter of smoke chimney, | 7 feet. |
| Height of smoke chimney above grates, | 50 " |
| Width of water-bottoms and all water-ways, | 6 inches. |
| Steam room in both boilers, above 12 inches above tubes, | 1150 cubic feet. |
| Weight of both boilers and all their appendages, except grate-bars and water, | 165,850 pounds. |
| Weight of grate-bars, | 15,120 " |
| Weight of water up to 12 inches above tubes, | 142,000 " |

The shell of the boilers, with the exception of their bottoms and the bottoms of the ash-pits where the iron is $\frac{7}{16}$-inch thick, is composed of ⅜-inch iron plate double riveted and braced for a hydrostatic test pressure of 50 pounds per square inch. The tube plates are of ½-inch thick iron, and the tubes are expanded around and riveted over them.

| | |
|---|---|
| Ratio of heating to grate surface, | 30·085 to 1·000. |
| Ratio of grate surface to calorimeter at back end of tubes, | 5·362 " |
| Ratio of grate surface to calorimeter at front end of tubes, | 7·150 " |
| Ratio of grate surface to cross area of chimney, | 6·503 " |

## SCREW.

One brass screw. The pitch expands both radially from hub to periphery, and fore and aft from the front to the back edge of the blades. At the hub the initial pitch is 22 feet and the final one 23 feet: at the periphery the initial pitch is 25 feet and the final one 26 feet. The corners of the blades at the periphery are rounded off on a radius of 2 feet on the developed surface. The screw is arranged to hoist up through a well in the usual manner. The hub is a frustrum of a sphere of 42 inches diameter: it is hollow with a thickness of metal of 2 inches in every place, except where it is joined by the blades, and there the metal is 4½ inches thick.

| | |
|---|---|
| Diameter of the screw, | 14½ feet. |
| Diameter of the hub, | 3½ " |
| Mean pitch in function of surface and propelling efficiency of the same, | 24·7 " |
| Length of the screw in the direction of its axis at the hub, | 3 " |
| Extreme length of the screw in the direction of its axis at radius of 5½½ feet, | 3⅜ " |
| Number of blades, | 2. |
| Mean fraction employed of the pitch in function of surface and propelling efficiency of the same, | 0·285 |
| Thickness of the blade above fillet at the hub, | 5½ inches. |
| Helicoidal area of the two blades, | 58·75 square feet. |
| Projected area of the two blades on a plane at right angles to axis, | 44·14 " |
| Diameter of the forward journal of the hub, | 18 inches. |
| Length of the forward journal of the hub, | 14 " |
| Diameter of the after journal of the hub, | 14 " |
| Length of the after journal of the hub, | 14 " |
| Weight of the screw, | 13,500 pounds. |
| Weight of the guides and chairs, | 2,869 " |
| Weight of the hoisting gear, viz: pillow blocks and saddles, | 4,005 " |

The total weight of the entire machinery, including spare pieces, but excluding the water in the boilers, is 537,265 pounds or 240 tons: inclusive of the water in the boilers it is 303⅜ tons.

## MAXIMUM PERFORMANCE IN SMOOTH WATER UNINFLUENCED BY WIND OR CURRENT.

The following is the maximum performance that, uninfluenced by wind or current, can be permanently sustained in smooth water. The different pressures in the cylinders are the mean of a collation of a large number of indicator diagrams.

| | |
|---|---|
| Vessel's draught of water, in feet and inches, | 15 6 |
| Vessel's greatest immersed transverse section, in square feet, | 551· |
| Vessel's displacement, in tons, | 2532· |
| Vessel's speed per hour, in geographical miles of 6086 feet, | 9·19 |

| | |
|---|---|
| Number of double strokes of engines' pistons and of revolutions of the screw made per minute, | 51· |
| Slip of the screw in per centum of its speed, | 26· |
| Fraction of the stroke of the piston at which the steam is cut off in the cylinders, | $\frac{1}{3}$ |
| Vacuum in the condenser, in inches of mercury, | 27. |
| Portion of throttle-valve open, | Wide. |
| Temperature of the feed-water, in degrees Fahrenheit, | 137. |
| Steam pressure in the boilers above the atmosphere, in pounds per square inch, | 18·0 |
| Steam pressure in the cylinders at the commencement of the stroke of the piston, in pounds per square inch above zero, by indicator, | 29·5 |
| Steam pressure in the cylinders at the point of cutting off, in pounds per square inch above zero, by indicator, | 27·0 |
| Steam pressure in the cylinders at the end of the stroke of the piston, in pounds per square inch above zero, per indicator, | 8·7 |
| Steam pressure in the cylinders against the piston during its stroke, in pounds per square inch above zero, per indicator, | 3·5 |
| Mean gross effective pressure on the piston in pounds per square inch, during its stroke, | 14·3 |
| Mean total pressure on the piston in pounds per square inch, during its stroke, | 17·8 |
| Gross effective horses power developed by the engines, | 705·662 |
| Total horses power developed by the engines, | 878·377 |
| Pounds of first quality anthracite consumed per hour, | 2777· |
| Pounds of refuse, in ashes, clinker, and dust per hour, | 397· |
| Pounds of combustible consumed per hour, | 2380· |
| Per centum of refuse from the anthracite, | 14·33 |
| Pounds of anthracite consumed per hour per square foot of grate, | 11·097 |
| Pounds of combustible consumed per hour per square foot of grate, | 9·515 |
| Pounds of anthracite consumed per hour per gross effective horse power, | 3·935 |
| Pounds of combustible consumed per hour per gross effective horse power, | 3·373 |
| Pounds of anthracite consumed per hour per total horse power, | 3·161 |
| Pounds of combustible consumed per hour per total horse power, | 2·709 |

## DRISTRIBUTION OF THE POWER DURING THE ABOVE PERFORMANCE.

The pressure required to work the engines and shafting *per se* being taken at 1½ pounds per square inch of piston, the power thus absorbed is 74·02 horses.

Deducting from the total power of 705·66 horses developed by the engines this power of 74·02 horses, there remains the power of 631·64 horses applied to the shaft, of which 7½ per centum or 47·37 horses are absorbed by the friction of the load.

The power expended in overcoming the cohesive resistance of the water by the screw blades, calculated in the ratio of the square of the velocity, and for a value of 0·45 pound per square foot of helicoidal surface moving in its helical path with a velocity of 10 feet per second, amounts to 40·60 horses.

The powers (47·37 and 40·60 horses) absorbed by the friction of the load and expended in overcoming the cohesive resistance of the water by the screw blades, being deducted from the power (631·64 horses)

applied to the shaft, there remains 543·67 horses power expended in the slip of the screw and in the propulsion of the hull. And as the slip of the screw is 26 per centum of its speed, the power expended in it is ($543{\cdot}67 \times {\cdot}26 =$) 141·35 horses, leaving ($543{\cdot}67 - 141{\cdot}35 =$) 402·32 horses expended in the propulsion of the simple hull.

Collecting the foregoing we have the following distribution of the power, namely:—

| | Horses power. | Per centum. |
|---|---|---|
| Gross indicator power developed by the engines, | 705·66 | |
| Power required to work the engines and shafting *per se*, | 74·02 | |
| Net power applied to the shaft, | 631·64 | or 100·00 |
| Power absorbed by the friction of the load, | 47·37 | " 7·50 |
| Power expended in overcoming the cohesive resistance of the water by the screw blades, | 40·60 | " 6·43 |
| Power expended in the slip of the screw, | 141·35 | " 22·38 |
| Power expended in the propulsion of the vessel, | 402·32 | " 63·69 |
| Totals, | 631·64 | or 100·00 |

## THRUST OF THE SCREW.

The thrust of the screw during the preceding maximum performance can be determined from the above "Distribution of the Power."

The power required to propel the simple hull is therein found to be 402·32 horses, equal to ($402{\cdot}32 \times 33000 =$) 13276560 pounds raised one foot high per minute. The speed of the vessel was 9·19 geographical miles of 6086 feet per hour, or $\left(\frac{9{\cdot}19 \times 6086}{60} =\right)$ 932·17233 feet per minute; the resistance of the vessel at this speed, or its equivalent the thrust of the screw, was consequently $\left(\frac{13276560{\cdot}}{932{\cdot}17233} =\right)$ 14243 pounds.

## PERFORMANCE AT SEA UNDER THE CONDITIONS OF ORDINARY PRACTICE.

In the two following tables will be found Abstracts of the Steam Logs of the "Brooklyn." The first gives the performance under steam alone; the second under steam and square sails combined. In these tables will be found the whole of her performance during her first cruise, which extended through the years 1859, 1860, and 1861, and was upon the Atlantic and Gulf of Mexico coasts of the United States. The tables include all that is recorded of her performance where the data is complete. Of course a considerable amount of steaming for short distances is excluded; and also some during which the log is imperfect.

The number of consecutive hours on each line of the tables, is the time during which the operation of the machinery, and the circumstances of wind, sea, and sail continued nearly uniform. The speed is the mean that was obtained by the chip log hove hourly by the officer of the deck, and the course of the vessel, direction and force of the wind, and sail set, are as recorded by him. The state of the sea is as recorded by the Engineer of the watch.

The number of revolutions of the screw and of double strokes of the engines' pistons made per minute,

were taken by a counter, and entered in the steam log at the end of each hour by the engineer of the watch. He also made hourly entries of the steam pressure in the boilers; vacuum in the condenser; proportion of throttle-valve open; pounds of anthracite consumed, and pounds of refuse derived from it; together with the various temperatures, &c., necessary to be known in forming a correct estimate of the performance of the machinery. The quantities given in the tables are the mean of this record for the consecutive hours of the steaming.

The coal consumed was Pennsylvania Anthracite of fair quality and burned with a very moderate rate of combustion. There was no forcing of the fires and the throttle-valve was habitually carried about half open. There was a slight tendency to priming with a wide throttle which was thus corrected.

The feed-water in its passage to the boiler from the hot-well was heated in a "heater" by means of the brine blown off from the boilers to maintain the concentration of their water at 1⅔ time that of the sea water. A continuous "surface blow" being used and the boiler pressure maintained in the heater, the feed-water which left the hot-well at the temperature of 110° Fahr., was subjected to the action of the boiler water with a temperature of 254° Fahr., and thence derived an additional temperature of 27° Fahr., making its temperature when entering the boiler 137° Fahr.

The slip of the screw is the difference between the product of its pitch and the number of its revolutions in a given time, and the speed of the vessel in the same time, expressed in per centum of the former.

When leaving port with 360 tons of coal on board, and all other weights full, the vessel's draught of water was 16 feet fore and aft.

The average draught of water for the whole steaming done was 15 feet 6 inches, the greatest immersed transverse section corresponding to which is 551 square feet, and the displacement 2532 tons.

## ABSTRACT OF THE STEAM LOG OF THE U. S. SCREW SLOOP "BROOKLYN," EMBRACING ALL HER PERFORMANCE UNDER STEAM ALONE.

| DATE | Number of consecutive hours. | Course of the Vessel. | WIND. Direction | WIND. Kind. | STATE OF THE SEA. | Speed of the Vessel per hour, in geographical miles of 6086 feet. | Number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute. | Steam pressure in Boilers, in lbs per sq. inch above atmosphere. | Vacuum in Condensers, in inches of mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Anthracite consumed per hour. | Per centum of Coal lost, in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Feb. 5 & 6, 1859. | 19 | S. E. | Abeam. | Light airs. | Smooth. | 7·000 | 41·88 | 16·8 | 25·5 | 0·88 | 80·44 | 1658 | 16⅓ |
| " 10, | 9 | W. by N. | N. W. | " breeze. | " | 6·889 | 89·83 | 17·1 | 25·( | 0·25 | 28·97 | 1447 | " |
| " 20, 21, & 22, | 59 | S. E. by S. | S. | Gentle " | " | 6·950 | 42·15 | 16·0 | 24·7 | 0·50 | 82·29 | 1956 | 14 |
| " 24, | 24 | S. E. | S. E. | " " | " | 7·166 | 88·58 | 19·1 | 24·5 | 0·40 | 28·73 | 1575 | " |
| March 1 & 2, | 27 | . . | Ahead. | Light " | " | 7·300 | 46·01 | 19·8 | 25·2 | 0·44 | 34·86 | 1861 | 15⅓ |
| " 25 & 26, | 13 | N. N. W. | N. W. | Strong " | Rough. | 6·154 | 41·28 | 21·1 | 25·4 | 0·37 | 88.78 | 1746 | 22 |
| April 13 & 14, | 17 | . . | On bow. | Light " | Smooth. | 6·176 | 87·35 | 22·3 | 25·7 | 0·84 | 82·08 | 1338 | 13¾ |
| May 5, | 10 | W. N. W. | N. N. W. | " airs. | " | 8·300 | 47·20 | 17·8 | 25·8 | 0·70 | 27·80 | 1790 | 18¾ |
| " 12 & 13, | 24 | S. E. | E. | Mod breeze. | Moderate. | 7·250 | 46·88 | 18·1 | 25·8 | 0·75 | 86·50 | 2060 | " |
| " 13 & 14, | 16 | S. E. | E. | Gentle " | Smooth. | 7·625 | 50·92 | 20·9 | 25·7 | 0·62 | 38·51 | 2088 | " |
| June 5, | 4 | S. E. | E. by N. | Light airs. | " | 8·500 | 51·80 | 24 0 | 25·5 | 0·50 | 82·61 | 2200 | 14 |
| " 7, | 4 | N. W. | S. E. | " " | " | 9·500 | 53·00 | 25·5 | 25·0 | 0 50 | 26·39 | 2175 | " |
| " 26, | 5 | S. E. | S. E. | Strong breeze. | Rough. | 6·800 | 47·84 | 28·0 | 26·0 | 0·62 | 41·68 | 1875 | 18 |
| " 30, | 3 | S. E. | S. | Light " | Smooth. | 8·333 | 50·67 | 20·0 | 27·0 | 0·62 | 32·46 | 1980 | " |
| July 4, | 3 | N. W. | N. N. W. | Mod. " | Moderate. | 8·000 | 50·20 | 22·0 | 26·0 | 0·62 | 34·55 | 2090 | 18 |
| " 12, 13, 14, & 15, | 61 | . . | On bow. | Light " | Smooth. | 6·623 | 87·10 | 16·8 | 26·5 | 0·3( | 26·69 | 1421 | 16 |
| Aug. 11, | 5 | S. by W. | S. W. | " " | Gentle swell. | 8·800 | 49·60 | 24·0 | 26·7 | 0·50 | 27·14 | 2200 | 21¾ |
| " 12, | 18 | S. S. W. | S. | " " | Smooth. | 7·778 | 44·87 | 14·0 | 26·9 | 0·50 | 28·81 | 2212 | " |
| " 13, | 18 | S. S. W. | S. | Mod. " | Moderate. | 7·055 | 45·00 | 19·0 | 27·0 | 0·50 | 35·68 | 2116 | " |
| " 14, | 24 | S. S. W. | S. by E. | Light " | " | 7·625 | 42·92 | 18·3 | 27 0 | 0·44 | 27·04 | 1964 | " |
| " 16 & 17, | 16 | S. E. | S. E. | Gentle " | Smooth. | 7·812 | 46·16 | 22·2 | 27.0 | 0·44 | 80·50 | 1982 | " |
| Sept. 5, | 12 | N. E. | N. W. | Light " | " | 7·833 | 42·67 | 22·0 | 27·0 | 0·31 | 24·61 | 1770 | 19¼ |
| " 23 & 24, | 48 | N. N. E. | . . | Calm. | " | 8·750 | 45·43 | 21·1 | 27·9 | 0·44 | 20·90 | 2183 | 26¾ |
| Nov. 10, | 12 | S. | S. | Strong breeze. | Rough. | 6·250 | 43·67 | 19·0 | 27·5 | 0·88 | 41·28 | 2091 | 17 |
| " 13, | 19 | S. | S. | Mod. gale. | " | 5·210 | 42·21 | 19·7 | 28·0 | 0·56 | 49·80 | 2084 | " |
| Dec. 17 & 18, | 20 | N. N. E. | N. E. | Light airs. | " | 8·450 | 47·58 | 19·7 | 26·8 | 0·50 | 27·07 | 2248 | 18½ |
| " 25, | 9 | S. W. by S. | S. | Strong breeze. | Smooth. | 7·778 | 46·22 | 18·4 | 26·7 | 0·50 | 80·90 | 2100 | 16⅓ |
| " 27 & 28, | 28 | S. S. W. | S. by E. | Mod. " | Moderate. | 7·964 | 44·07 | 16·4 | 26·8 | 0·50 | 25·79 | 1972 | " |
| Jan. 4 & 5, 1860. | 23 | S. E. by S. | On bow. | Light " | Smooth. | 9·044 | 46·87 | 17·8 | 26·8 | 0·50 | 20·00 | 2026 | 8 |
| " 26, 27, & 28, | 51 | N. E. | N. N. E. | Gentle " | Moderate. | 7·060 | 89·68 | 16·8 | 26·8 | 0·38 | 26·94 | 2048 | 15⅓ |
| Feb. 21, | 24 | S. E. | S. E. | Mod. " | Smooth. | 6·625 | 42·46 | 15·8 | 26·7 | 0·50 | 85·98 | 2185 | 20 |
| " 25 & 26, | 36 | N. N. E. | N. N. W. | Strong " | Rough. | 7·194 | 44·81 | 18 5 | 26·8 | 0·50 | 34·07 | 2236 | " |
| March 12, | 12 | S. W. | S. W. | Gentle " | Smooth. | 7·333 | 43·23 | 16·0 | 26·5 | 0·50 | 80·34 | 2185 | 19⅓ |
| " 15, | 10 | S. E. | E. | Light " | " | 8·600 | 47·80 | 17·6 | 26·9 | 0·50 | 26·12 | 2122 | " |
| " 18, | 12 | S. S. W. | S. W. | Mod. " | Rough. | 7·333 | 48·80 | 16·3 | 26·7 | 0·50 | 80·45 | 2242 | " |
| April 30, | 24 | E. N. E. | N. N. E. | Gentle " | Smooth. | 7·383 | 42·18 | 15·6 | 26·8 | 0·50 | 28·60 | 2109 | 14⅔ |
| May 1 & 2, | 35 | N. E. | E. N. E. | Mod. " | Rough. | 5·912 | 40·97 | 16·5 | 26·8 | 0·50 | 40·75 | 2125 | 8 |
| June 3 & 4, | 44 | E. by N. | E. | Gentle " | Moderate. | 6·886 | 42·91 | 15·7 | 26·8 | 0·50 | 84·09 | 2131 | 9 |
| July 11, 12. 13, 14, 15, | 89 | N. N. E. | N. E. | Light " | Smooth. | 8·798 | 46·72 | 18·0 | 26·9 | 0·50 | 22·67 | 2237 | 7 |
| " 27 & 28, | 48 | E. S. E. | S. E. by E. | Mod. " | Moderate. | 6·875 | 43·88 | 17·0 | 26·9 | 0·50 | 84·92 | 2354 | 19 |
| Aug. 14, | 24 | S. | S. E. | " " | " | 7·209 | 44·14 | 16·9 | 26·8 | 0·50 | 82·98 | 2277 | 21¼ |
| " 17, 18, & 19, | 72 | S. | S. | Light " | " | 8·347 | 47·25 | 17·9 | 26·7 | 0·50 | 27·45 | 2252 | " |
| " 22 & 23, | 25 | S. S. W. | S. E. | Light airs. | Smooth. | 7·720 | 42·14 | 15·5 | 26·7 | 0·40 | 24·78 | 2085 | " |
| Sept. 7, | 4 | E. by N. | E. | Gentle breeze. | " | 7·750 | 47·30 | 19·1 | 26·9 | 0·50 | 30·38 | 2240 | 16 |
| " 11, | 7 | E. by N. | E. N. E. | " " | " | 8·286 | 47·20 | 17·4 | 26 9 | 0·53 | 27·92 | 2240 | 10 |
| " 16, | 13 | W. S. W. | W. N. W. | Light " | " | 8·077 | 45·85 | 16·8 | 26·8 | 0·50 | 26·81 | 2286 | " |
| Oct. 16 & 17, | 12 | E. by N. | N. E. | " " | " | 8·667 | 44·84 | 14·6 | 26·9 | 0·50 | 19·78 | 1955 | 11½ |
| " 22, 23, 24, & 25, | 66 | W. by S. | W. | Gentle " | Rough. | 5·955 | 40·29 | 17·5 | 26·9 | 0·50 | 82·52 | 2104 | 16¾ |
| Nov. 6 & 7, | 11 | E. by N. | E S. E. | Light " | Smooth. | 8·454 | 48·22 | 19·5 | 26·7 | 0·56 | 28·00 | 2026 | 20 |
| " 12 & 13, | 15 | W. by S. | W. | " " | Gentle. | 8·200 | 48·57 | 20·1 | 26·7 | 0·56 | 30·67 | 2182 | " |
| " 17, 18, & 19, | 60 | N. | N. N. E | Gentle " | Moderate. | 7·466 | 45·98 | 18·8 | 26·8 | 0·58 | 83·25 | 2240 | " |
| " 22 & 23, | 42 | N. E. | N. E. by E. | " " | " | 7·738 | 46·77 | 18·2 | 26·8 | 0·50 | 82·06 | 2184 | " |
| Jan. 9, 1861. | 9 | S. S. E. | S. S. E. | Light " | Smooth. | 7·889 | 45·80 | 19·1 | 26·7 | 0·50 | 29·27 | 1941 | 17¼ |
| " 10, | 9 | S. | S. W. | Strong " | Rough. | 6·333 | 41·98 | 16·9 | 26·6 | 0·50 | 38·04 | 2078 | " |
| " 25, | 10 | S. E. | S. E. | Light " | Moderate. | 7·800 | 44·81 | 16·4 | 26·6 | 0·50 | 28·56 | 2070 | 16½ |
| " 27, | 8 | S. W. by S. | W. S. W. | Fresh " | Heavy. | 4·500 | 37·73 | 15·8 | 26·3 | 0·38 | 51·01 | 1984 | " |
| Feb. 3, | 8 | N. W. | S. | Light airs. | Smooth. | 8·750 | 46·21 | 16·7 | 26·9 | 0·50 | 22·25 | 2072 | 17¼ |
| " 4 & 5, | 32 | N. W. by N. | N. | Mod. breeze. | " | 7·125 | 48·23 | 19·1 | 27·0 | 0·50 | 32·82 | 1985 | " |
| March 22 & 23, | 33 | S. E. | S by. E. | " " | " | 5·485 | 40 73 | 15·7 | 26·6 | 0·50 | 44·70 | 2058 | 16⅔ |
| " 25, | 20 | N. E. | E. by N. | " " | Moderate. | 4·850 | 39·27 | 16·5 | 26·9 | 0·50 | 49·28 | 2027 | " |
| May 25 & 26, | 18 | S. W. | S. W. | Light " | Smooth. | 6·000 | 44·00 | 17·4 | 26·3 | 0·50 | 44·00 | 2041 | 22 |
| Sept. 15, 16, & 17. | 72 | S. E. | E S. E. | Mod. " | Moderate. | 6·306 | 42·20 | 18·9 | 25·4 | 0·56 | 38·65 | 1964 | 20¾ |
| " 19 & 20, | 30 | N. N. E. | N. N. E. | Light " | Smooth. | 5·583 | 33·48 | 13·4 | 24·5 | 0·25 | 82·18 | 1423 | " |
| Means, | | | 25° from Ahead. | Gentle breeze. | Gentle swell. | 7·228 | 43·47 | 17·5 | 26·5 | 0·48 | 31·72 | 2039 | 17 |

## ABSTRACT OF THE STEAM LOG OF THE U. S. SCREW SLOOP "BROOKLYN," EMBRACING ALL HER PERFORMANCE UNDER STEAM AND SQUARE SAILS COMBINED.

| DATE. | Number of consecutive hours. | Course of the Vessel. | WIND. Direction. | WIND. Kind. | STATE OF THE SEA. | Speed of the Vessel per hour, in geographical miles of 6086 feet. | Number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute. | Steam pressure in Boilers, in pounds per sq. inch above the atmosphere. | Vacuum in Condensers, in inches of Mercury. | Proportion of Throttle-valve open. | Slip of the Screw, in per centum of its speed. | Pounds of Anthracite consumed per hour. | Per centum of the Coal lost, in ashes, clinker, and dust. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1859. | | | | | | | | | | | | | |
| Feb. 6, 7, & 8, | 56 | S. S. W. | On quarter. | Strong breeze. | Rough. | 9·018 | 44·00 | 17·8 | 25·0 | 0·40 | 15·88 | 2118 | $16\frac{1}{5}$ |
| " 9 & 10, | 34 | S. W. | N. W. | " " | " | 4·912 | 37·85 | 16·2 | 24·2 | 0·28 | 45·96 | 1490 | " |
| " 23, | 24 | S. | On bow. | Mod. " | Smooth. | 8·209 | 34·72 | 16·4 | 24·2 | 0·12 | 2·90 | 1409 | 14 |
| May 9 & 10, | 24 | N. N. W. | E. N. E. | Light " | Rough. | 8·625 | 47·69 | 16·4 | 25·4 | 0·75 | 25·73 | 1800 | $18\frac{3}{4}$ |
| " 13, | 4 | S S. E. | E. | Mod. " | Moderate. | 8·750 | 46·92 | 17·3 | 26·0 | 0·62 | 28·41 | 2025 | " |
| " 17, | 9 | N. W. | Abeam. | Strong " | Smooth. | 7·444 | 37·40 | 20·5 | 26·2 | 0·25 | 18·26 | 1925 | " |
| July 10 & 11, | 14 | N. N. E. | E. | Gentle " | " | 8 607 | 39·92 | 17·4 | 26·8 | 0·30 | 11·46 | 1380 | $21\frac{3}{4}$ |
| August 12, | 6 | S. S. W. | S. E. | Fresh " | " | 9·500 | 47·05 | 19·0 | 27·0 | 0·50 | [illegible]7·09 | 1800 | " |
| " 13, | 6 | S.S. W. | S. E. | Mod. " | Moderate. | 8·500 | 46·40 | 21·0 | 27·0 | 0·44 | 24·77 | 1815 | " |
| " 15, | 6 | S. S. W. | S. E. | " " | " | 7·667 | 35·80 | 11·5 | 27·0 | 0·81 | 12·05 | 1570 | " |
| Sept. 1, 2, 3, & 4, | 72 | N. N. E. | E. | " " | " | 8·611 | 48·74 | 21·5 | 27·0 | 0·37 | 19·16 | 1876 | $19\frac{1}{4}$ |
| " 20, 21, & 22, | 60 | N. by E. | Abeam. | Light " | Smooth. | 8·917 | 46·00 | 21·2 | 27·7 | 0·38 | 20·40 | 2110 | $26\frac{3}{4}$ |
| Nov. 8 & 9, | 32 | S. | E. S. E. | Strong " | Rough. | 8·281 | 46·48 | 20·9 | 27·5 | 0·38 | 26·76 | 2121 | 18 |
| " 10, 11, & 12, | 60 | S. S. W. | Forward beam. | Fresh " | Moderate. | 9·400 | 49·10 | 19·0 | 27·8 | 0·50 | 21·39 | 2221 | " |
| " 14, | 21 | S. W. | N. N. W. | Gale. | Rough. | 11·050 | 47·90 | 21·5 | 28·[illegible] | 0·38 | 5·26 | 2149 | " |
| " 17, 18, 19, 20, 21, | 99 | W. S. W. | Abeam. | Fresh breeze. | Moderate. | 10·282 | 48·06 | 18·1 | 26·[illegible] | 0·54 | 13·80 | 2259 | " |
| Dec. 15, 16, & 17, | 58 | N. N. E. | N. N. W. | " " | Rough. | 9·108 | 46·37 | 17·8 | 2[illegible] | 0·45 | 19·39 | 2188 | $18\frac{1}{2}$ |
| " 26 & 27, | 30 | S. W. by S. | S. by E. | " " | Moderate. | 9·038 | 40·37 | 16·5 | 2[illegible] | 0·32 | 8·11 | 1790 | $16\frac{1}{3}$ |
| 1860. | | | | | | | | | | | | | |
| Jan. 1 & 2, | 20 | N. E. | N. W. | Strong gale. | Heavy. | 3·950 | 33·33 | 18·0 | 6·8 | 0·25 | 51·32 | 1540 | 8 |
| " 23 & 24, | 32 | N. W. by N. | N. E. | Gentle breeze. | Smooth. | 9·500 | 44·49 | 14·7 | [illegible]·7 | 0·50 | 12·31 | 2166 | $15\frac{2}{3}$ |
| " 25, | 24 | N. E. | E. by S. | Strong " | Moderate. | 10·000 | 47·83 | 14·3 | 26·7 | 0·50 | 14·14 | 2218 | " |
| Feb. 19 & 20, | 36 | S. E. | E. by N. | Gentle " | Smooth. | 8·583 | 45 78 | 15·9 | 26·8 | 0·50 | 23·02 | 2184 | 20 |
| " 22, 23, & 24, | 72 | . . | On quarter. | Strong " | Moderate. | 10·514 | 47·00 | 15·4 | 26·8 | 0·50 | 8·13 | 2204 | " |
| " 26, | 5 | N. N. E. | S. | Mod. " | Smooth. | 10·000 | 50·90 | 20·5 | 26·9 | 0·56 | 19·32 | 2500 | " |
| Mar. 11, | 12 | S. S. W. | W. N. W. | Strong " | " | 9·833 | 47·27 | 17·2 | 26·5 | 0·50 | 14·58 | 2185 | $19\frac{1}{3}$ |
| " 16, | 24 | S. | E. by S. | Gentle " | " | 9·458 | 47·28 | 14·6 | 26·8 | 0·50 | 17·90 | 2228 | " |
| " 17 & 18, | 36 | S. W. | S. E. | Gale. | Rough. | 9·167 | 43·40 | 14·2 | 26·7 | 0·41 | 13·28 | 2127 | " |
| " 19 & 20, | 34 | S. by W. | W. | Fresh breeze. | Moderate. | 10·206 | 46·37 | 16·[illegible] | 26·6 | 0·50 | 9·62 | 2204 | " |
| " 22, 23, 24, 25, 26, | 112 | W. by S. | Abeam. | Gentle " | Gentle. | 9·411 | 44·72 | 16·0 | 26·8 | 0·50 | 13·59 | 2207 | $14\frac{2}{3}$ |
| " 27 & 28, | 32 | . . | On bow. | Gale. | Heavy. | 4·719 | 32·16 | 20·1 | 26·7 | 0·25 | 39·74 | 1984 | " |
| " 28, | 6 | . . | Abeam. | Fresh breeze. | Rough. | 10·500 | 47·70 | 18·1 | 26·8 | 0·50 | 9·60 | 2223 | " |
| May 10, | 12 | S. W. | N. W. | " " | Moderate. | 9·417 | 44·68 | 17·4 | 26·8 | 0·50 | 13·37 | 2125 | 8 |
| June 25 & 26, | 18 | S. E. | Forward beam. | Light " | Smooth. | 8·722 | 41·35 | 15·5 | 26·7 | 0·40 | 13·39 | 1911 | 9 |
| July 26, | 24 | S. E. | E. | Gentle " | " | 8.667 | 43·65 | 14·4 | 26·9 | 0·50 | 18·48 | 2189 | 19 |
| " 29, 30, & 31, | 72 | N. N. E. | Abeam. | Strong " | Moderate. | 9·764 | 47·71 | 18·3 | 26·9 | 0·50 | 15·98 | 2345 | " |
| August 15 & 16, | 48 | S. by E. | " | Gentle " | " | 8·688 | 46·05 | 16·2 | 26·7 | 0·50 | 22·52 | 2252 | $21\frac{1}{4}$ |
| " 20, | 9 | S. by W. | S. E. | Mod. " | Smooth. | 10·222 | 48·14 | 15·4 | 26·8 | 0·50 | 12·79 | 2240 | " |
| Nov. 20 & 21, | 48 | N. N. W. | N. N. E. | Light " | Moderate. | 9·563 | 47·68 | 18·7 | 26·7 | 0·52 | 17·63 | 2240 | 20 |
| " 23, | 6 | N. | S. W. | " " | " | 9·833 | 47·52 | 18·7 | 26·9 | 0·50 | 15·04 | 2184 | " |
| " 25, | 24 | N. E. | N. N. W. | Gentle " | Rough. | 7·208 | 35·09 | 13·6 | 27·[illegible] | 0·12 | 15·65 | 1671 | " |
| " 26 & 27, | 40 | N. N. E. | On quarter. | Mod. " | Moderate. | 10·150 | 45·41 | 13·7 | 26·9 | 0·50 | 8·21 | 2184 | " |
| 1861. | | | | | | | | | | | | | |
| Jan. 10 & 11, | 28 | S. W. | N. N. W. | Strong " | Rough. | 9·643 | 44·68 | 17·2 | 26·7 | 0·47 | 11·37 | 2008 | $17\frac{1}{4}$ |
| " 12, 13, & 14, | 35 | . . | Abeam. | Gentle " | Moderate. | 9·543 | 45·16 | 16·8 | 26·8 | 0·50 | 13·21 | 2088 | " |
| " 25, | 14 | S. S. E. | Abaft beam. | Light " | " | 8·000 | 45·97 | 18·0 | 26·7 | 0 50 | 28·53 | 2053 | $16\frac{1}{2}$ |
| " 26, | 7 | S. by W. | E. | Fresh " | Heavy. | 8·143 | 43·73 | 15·0 | 26·6 | 0·50 | 28·54 | 2070 | " |
| " 27 & 28, | 40 | S. by W. | N. W. by N. | Mod. " | Rough. | 8·225 | 42·79 | 15·2 | 26 6 | 0·50 | 21·06 | 2[illegible]84 | " |
| " 29 & 30, | 44 | W. by S. | On quarter. | Light " | Smooth. | 9·070 | 46·17 | 16·7 | 26·7 | 0·50 | 19·38 | 2138 | " |
| Feb. 2, | 9 | N. W. | S. E. | " " | " | 9·333 | 48·95 | 18·8 | 26·6 | 0·50 | 21·70 | 2016 | $17\frac{1}{4}$ |
| " 16 & 17, | 40 | . . | Forward beam. | Fresh " | Heavy. | 5·275 | 34·73 | 15·6 | 27·0 | 0·25 | 37·63 | 1439 | 21 |
| Mar. 24 & 25, | 28 | S. E by S. | E. by N. | Gentle " | Smooth. | 8·464 | 44·75 | 14·8 | 26·9 | 0·50 | 22·32 | 2128 | $16\frac{2}{3}$ |
| " 29, 30, & 31, | 52 | N. W. | N. E. | Mod. " | Moderate. | 9·173 | 46·88 | 16·1 | 26·9 | 0·50 | 19·65 | 2240 | " |
| Sept. 18 & 19, | 36 | . . | On bow. | Light " | Smooth. | 6·555 | 37·14 | 13·7 | 23·5 | 0·37 | 27·52 | 1531 | $20\frac{3}{4}$ |
| Means, | | | 87° from Ahead. | Mod. breeze. | Moderate. | 8·880 | 44·44 | 17·1 | 26·6 | 0·45 | 18·87 | 2062 | 18 |

## RESULTS OF THE STEAM LOG.

The following table contains a Synopsis of the two preceding tables of the Steam Log of the "BROOKLYN," and the economic and potential results deduced therefrom. The first column contains the average results of the steaming done under steam alone; the second column contains the average results of the steaming done under steam and square sails combined; the third column contains the mean of the preceding two.

The steam pressures in the cylinder per indicator, are the means from a careful analysis of one hundred and twenty-six sets of diagrams, four diagrams to a set, taken daily during the steaming under average conditions, and truly exhibiting the distribution of pressure in the cylinder corresponding with the average of the observations of the steam log. It will be observed that these pressures are given above the zero line or line of no pressure: the mean pressure of the atmosphere being taken at 14·7 pounds per square inch, their relation to the atmospheric line can be easily ascertained. The "Mean Gross Effective Pressure" represents the area of the indicator diagram; it includes the pressure required to work the engines *per se*, but excludes the back pressure against the pistons. The "Total Pressure" is the sum of the mean gross effective pressure and the back pressure against the piston. The total horses power developed by the engines, computed from this pressure, expresses the entire dynamic effect of the steam, inclusive of moving the load, working the engines, and overcoming the back pressure against the pistons.

Careful experiments made by means of a tank on boilers of the type and proportions of the "BROOKLYN'S," and with the same kind of coal and per centum of refuse, show their evaporation to average, under the ordinary conditions of firing at sea, 9½ pounds of water from a temperature of 137° Fahr., by one pound of anthracite. Computing, now, the weight of steam discharged per hour from the cylinders,—by means of the pressure at the end of the stroke of the piston and FAIRBAIRN'S formula; and adding to it the weight of steam condensed in the cylinder to produce the total power developed by the engines, according to JOULE'S equivalent; and the weight of steam that would have been evaporated had the heat been so applied which was lost in blowing off to maintain the concentration of the water in the boilers at 1⅔ time the natural concentration of sea-water; we find the difference between the sum and the weight due to an evaporation of 9½ pounds of water per pound of coal, to be 20·2 per centum of the latter. This difference expresses the losses by leakages of all kinds, errors of observation and data, and condensation in the cylinders due to all other causes except the production of the power. The loss by "blowing-off" alone amounts to 12·902 per centum of the total evaporation.

As the total evaporation was 19,484·500 pounds of water per hour, and as the condensation in the cylinder to produce the total power developed by the engine was 2,513·890 pounds of steam, it follows that of the total heat put into the steam $\left(\frac{2{,}513{\cdot}890 \times 100}{19{,}484{\cdot}500} = \right)$ 12·9 per centum was utilized or converted into mechanical power. In all the calculations REGNAULT'S tables of the temperature, total, and latent heat of steam have been used.

It will be observed that the steam entered the cylinders at 8 pounds less pressure than the boiler pressure; of this reduction at least 5 pounds are due to throttle-valve having been nearly half closed.

The back pressure in the condenser was 1·7 pound per square inch above zero, and in the cylinders 3·7 pounds per square inch.

In estimating the performance of the machinery, care must be taken to discriminate between the results due to mal-proportion, and those normal to the system when properly proportioned.

The total pressure on the pistons, we see, was 14·1 pounds per square inch; of this, the back pressure (3·7 pounds) and the pressure required to work the engines *per se* (estimated at 1·5 pound) amount to 5·2 pounds per square inch, leaving only (14·1 — 5·2 =) 8·9 pounds per square inch or $\left(\frac{8\cdot9 \; 10 \; 0}{14\cdot1} = \right)$ 63 per centum utilized. Had the area of the pistons been reduced about one-half, the total pressure would have been 28·2 pounds per square inch, while the sum of the back and friction pressures remaining about the same (5·2 pounds), there would have been utilized $\left(\frac{\overline{28\cdot2 - 5\cdot2} \times 100}{28\cdot2} = \right)$ 81½ per centum, being an addition to the economy of the fuel of 18½ per centum, besides the advantages of a smaller cylinder. The total indicated horse power was obtained at a cost of 3·287 pounds of coal per hour: with the cylinder of reduced area the cost of the mean gross effective horse power would have been about $\left(\frac{3\cdot287 \times 28\cdot2}{\overline{28\cdot2 - 3\cdot7}} = \right)$ 3·783 pounds of coal per hour. It was, under the actual conditions, 4·634 pounds.

To obtain, with the same measure of expansion, an average total pressure in the cylinders of 28·2 pounds per square inch, would require a boiler pressure of about 26 pounds per square inch above the atmosphere, —allowing 3 pounds for difference of pressure in the boilers and in the cylinders at the commencement of the stroke of the pistons. This pressure may safely be carried with the use of sea-water in the boilers.

We thus perceive that for a maximum economic result, supposing the boilers and the screw propeller to be properly proportioned to the vessel, the capacity of the cylinders was about double what it should have been. Or, if the capacity of the cylinders and the dimensions of the screw propeller be correct, then, the boilers are only half what they should be. Or, if the capacity of the cylinders and the boiler be proper, then, the dimensions of the screw propeller are not correct. Its pitch, in this latter case should be so much greater than it is that, with the same consumption of fuel, the average total pressure in the cylinders would be double; the number of revolutions of the screw in a given time being, of course, correspondingly less. Increasing the pitch, however, would correspondingly increase the slip of the screw, already too great, unless its diameter could also be increased, which, owing to the vessel's draught of water, is inadmissible.

The machinery of the "BROOKLYN" has the common fault of too much cylinder capacity, too little boiler, and too much slip of the screw. The details, however, have worked well and given great satisfaction.

## SYNOPSIS OF THE STEAM LOG OF THE U. S. SCREW SLOOP OF WAR "BROOKLYN."

| | Under Steam Alone. | Under Steam and Square Sails combined. | Mean of the two preceding Columns. |
|---|---|---|---|
| Total number of hours, | 1,535· | 1,694· | 3,229· |
| Kind of wind, | Gentle breeze. | Moderate breeze. | . . |
| Angle made from ahead by the wind with the line of the vessel's keel, | 25·° | 87·° | . . |
| State of the sea, | Gentle swell. | Moderate sea. | . . |
| Speed of the Vessel per hour, in geographical miles of 6086 feet, | 7·228 | 8·880 | 8·095 |
| Number of double strokes of Engines' Pistons, and of revolutions of the Screw, made per minute, | 43·47 | 44·44 | 43·98 |
| Slip of the Screw, in per centum of its axial speed, | 31·72 | 18·87 | 24·41 |
| Steam pressure in the boilers, in pounds per square inch above the atmosphere, | 17·5 | 17·1 | 17·3 |
| Vacuum in Condensers, in inches of mercury, | 26·5 | 26·6 | 26·55 |
| Proportion of Throttle-valve open, | 0·48 | 0·45 | 0·46 |
| Temperature in degrees Fahr. of the Hot-well, | 110·° | 110·° | 110·° |
| Temperature in degrees Fahr. of the Feed-water (after passing through the Heaters), | 137·° | 137·° | 137·° |
| Fraction of the Stroke of the Piston completed when the steam was cut off, per indicator, | ⅓ | ⅓ | ⅓ |
| Steam Pressure in Cylinder, per Indicator. — In pounds per square inch above zero at commencement of Stroke of Piston, | 24·0 | 24·0 | 24·0 |
| In pounds per square inch above zero at point of cutting off the Steam, | 19·3 | 19·3 | 19·3 |
| In pounds per square inch above zero at end of Stroke of Piston, | 6·9 | 6·9 | 6·9 |
| In pounds per square inch above zero against the Piston during its stroke, | 3·7 | 3·7 | 3·7 |
| Mean Gross Effective Pressure in pounds per square inch on Piston during its stroke, | 10·4 | 10·4 | 10·4 |
| Mean Total Pressure in pounds per square inch on Piston during its stroke, | 14·1 | 14·1 | 14·1 |
| Gross Effective Horses Power developed by the Engines, | 437·428 | 446·787 | 442·56 |
| Total Horses Power developed by the Engines, | 616·784 | 630·547 | 624·02 |
| Pounds of Anthracite consumed per hour, | 2,039· | 2,062· | 2,051· |
| Pounds of Refuse from Anthracite per hour, in ashes, clinker, and dust, | 347· | 371· | 360· |
| Per centum of Refuse from the Anthracite, | 17· | 18· | 17·5 |
| Pounds of Combustible consumed per hour, | 1,692· | 1,691 | 1,691· |
| Pounds of Anthracite consumed per hour per square foot of grate, | 8·148 | 8·289 | 8·196 |
| Pounds of Combustible consumed per hour per square foot of grate, | 6·761 | 6 757 | 6·757 |
| Pounds of Anthracite consumed per hour per Gross Effective Horse Power, | 4·661 | 4·615 | 4·634 |
| Pounds of Anthracite consumed per hour per Total Horse Power, | 3·306 | 3·270 | 3·287 |
| Pounds of Combustible consumed per hour per Gross Effective Horse Power, | 3·868 | 3·785 | 3·821 |
| Pounds of Combustible consumed per hour per Total Horse Power, | 2·748 | 2·682 | 2·710 |
| Evaporated from temperature of Feed-water. — Pounds of Steam discharged per hour from Cylinders into Condenser, calculated from the pressure of the steam at the end of the Stroke of the Piston, | . . | . . | 11,419·436 |
| Pounds of Steam per hour equivalent to the Heat annihilated in the Cylinders to produce the Total Power of the Engines, calculated from JOULE's equivalent, | . . | . . | 1,615·212 |
| Pounds of Steam that would have been evaporated per hour, had the Heat been so applied that was expended in "blowing off," to maintain the Sea-water in the Boilers at one and three-fourths time the natural concentration, supposing the Boilers to evaporate 9¼ pounds of water per pound of Anthracite, | . . | . . | 2,513·890 |
| Sum of the above three quantities, | . . | . . | 15,548·538 |
| Pounds of Water evaporated from temperature of Feed-water (137° Fahr.) per hour, supposing 9¼ pounds of water vaporized by one pound of Anthracite, | . . | . . | 19,484·500 |
| Per centum of the Steam evaporated in the Boilers not accounted for by the Indicator, and by "blowing off," being the per centum which the difference between the quantities on the two preceding lines is of the quantity on the preceding line, | . . | . . | 20·2 |

# EXPERIMENT

## TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER

OF

## THE UNITED STATES STEAMER

# "JACOB BELL,"

WITH BLACKHEATH ANTHRACITE.

2 F

# EXPERIMENT

## TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER

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## UNITED STATES STEAMER "JACOB BELL,"

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---

THE following experiment was made upon the boiler of the U. S. Steamer "JACOB BELL" to determine its evaporative efficiency with Blackheath Anthracite. The vessel lay alongside the wharf at the Washington Navy Yard and the evaporation was performed under the atmospheric pressure, an escape-pipe being fitted to the manhole at the centre of the top of the boiler shell to permit the free egress of the steam without entraining watery particles. The manhole was an oval with diameters of 16 and 18 inches, and the escape-pipe was of equal area.

The boiler was new and never had steam upon it before. The shell was double riveted and its perfect tightness had been assured under a hydrostatic pressure of 40 pounds per square inch. The entire exterior surface, including the steam chimney, was covered with No. 1 felt stitched to No. 1 canvass, over which was another covering of canvass thoroughly painted.

The escape-pipe on leaving the boiler was bent to an elbow, the lowest point of which was perforated with a hole to allow the discharge of the water of condensation in the pipe and thus prevent its return to the boiler.

The number of cubic feet of water evaporated was ascertained by measurement in a tank 72⅛ inches in length, 48 inches in height, and 24 inches in width. This tank was placed upon the wharf considerably above the level of the water in the boiler, and was filled by gravity from the reservoir of the Navy Yard through a hose fitted with a stop-cock at the tank. It was discharged by gravity into the boiler through another hose fitted with a stop-cock at the boiler. The entire apparatus was in open view and every precaution was taken against possibility of loss of water by leakage. A glass water-gauge attached to the boiler

showed the precise level of the water within, which was admitted as fast as evaporated, and the water level kept nearly constant at 8 inches above the top of the tubes.

The boiler having been filled to about 8 inches above the top of the tubes, a fire was made in the furnaces with pine wood until the water was raised to the boiling point; the wood was then allowed to burn down to the few coals required for kindling the anthracite,—and the time, and the exact height of water in the glass gauge having been noted,—the anthracite was fired, and the experiment began. During its continuance every pound of the coal was accurately weighed out hourly, and burned with as much uniformity as possible; the weight varying from 500 to 600 pounds per hour. Care was taken to maintain the fires at the constant thickness of 8 inches, and the water level at the same height on the gauge. The cleaning of the fires was performed at regular intervals and the whole of the refuse in ashes, clinker, &c., was carefully weighed in the *dry* state. At the close of the experiments the fires were burned out as nearly as possible, the furnaces drawn at the moment the time expired, and the unconsumed coal picked out, weighed, and deducted. At the expiration of the experiment, the water in the glass gauge was brought precisely at the same point as at the commencement.

The experiment continued uninterruptedly seventy-two hours, during which it was supervised by three Assistant Engineers of the Navy, who stood regular watches of four hours and attended personally to the weighing and measuring. They also kept a tabular record in which was noted hourly the height of the barometer; temperatures of the atmosphere in the shade, of the fire-room, and of the feed-water in the tank; the exact weight of coal fired: and the time at which each tankful of water was emptied. The ashes were weighed as withdrawn and the amount entered in the record at the time. The totals and means of these quantities will be found in the table hereinafter given of the results of the experiment.

The dampers were never closed, and the rate of combustion, though not at all forced, was the maximum naturally determined by the conditions of the boiler in connexion with a fire bed of 8 inches thick. Great care was taken to keep this bed uniform in condition and free from heaps or holes. The coal was not picked, but was taken in the condition delivered by the contractors; it was in lumps of uniform, but rather large size, and free from dust. Its quality was good. The air-holes in the furnace doors were kept open during the entire experiment.

## DESCRIPTION OF THE BOILER.

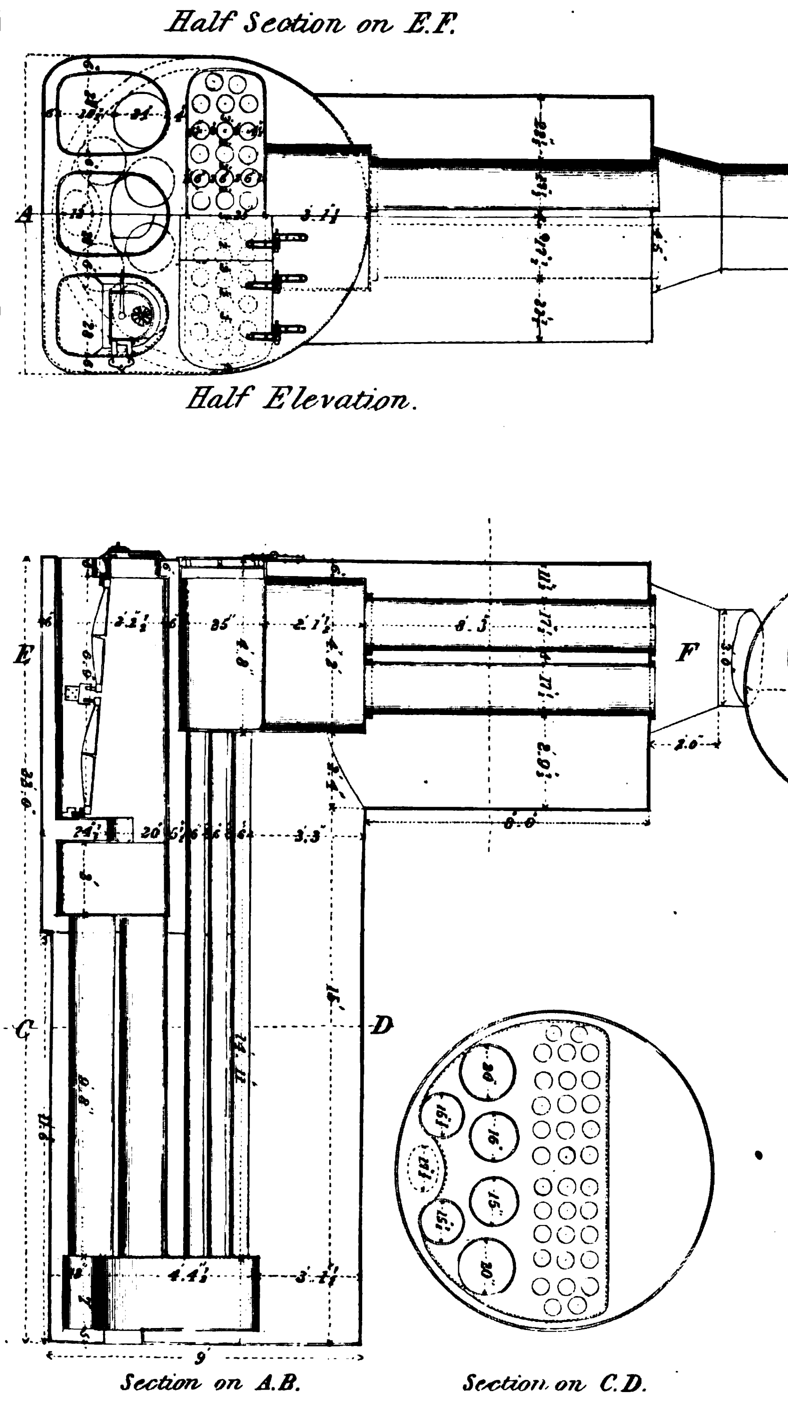
The shell of the boiler consists of a front rectangular in plan and extending 10 feet 6 inches to the back of the smoke connexion behind the bridge walls; thence, for the remaining 11 feet 6 inches, it is a cylinder of 9 feet diameter. This front is 9 feet 3 inches high, and semicircular on top, the semicircular part being an extension of the upper half of the cylinder behind: the offset of 3 inches is at the bottom.

The furnaces are three in number and semicircular on top. The angles of the ash-pits are rounded on a radius of 6 inches. The bridge walls are 8 inches wide. The smoke connexion immediately behind the bridge walls is in common for all three furnaces; its top is flat and on a level with the crown of the furnaces; its sides and bottom are also flat. From this smoke connexion six circular flues extend horizontally to the smoke connexion at the back end of the boiler which, like the smoke connexion behind the bridge walls, is in common for all the furnaces; in section it is concentric with the circular shell of the boiler except on top, where it is flat; water space between connexion and shell 4½ inches wide. From this smoke connexion at the back of the boiler there is returned horizontally, above the lower flues and the fur-

Plate VIII.

Boiler of U.S. Steamer

"JACOB BELL."



naces, three rows, in height, of tubes of 6 inches external diameter, which discharge at the front of the boiler into a capacious uptake. The lower part of this uptake is rectangular and 25 inches high; the upper part is cylindrical, 50 inches in diameter and 37½ inches in height. From the top of the uptake arise, vertically, four superheating flues through which pass the products of combustion: these flues are surrounded by a steam chimney, and discharge into the smoke chimney surmounting them.

In the spandrels of the furnaces there are manholes, and in the spandrels of the ash-pits handholes. The furnace doors are semicircular on top, 16 inches in width of opening and 18 inches in extreme height of opening; they are fitted with air registers, and lined with a wrought iron plate perforated with holes of ¼-inch diameter for the distribution of the air. Area of air admission through each door 18 square inches. The lining plate is so arranged that all the air passes through the holes, which are separated by spaces ¼-inch wide.

The spaces in the clear between the six inches diameter tubes, both vertically and horizontally, are two inches.

The shell of the boiler is double riveted and strongly stayed. The bottom of the rectangular front and the bottom of the ash-pits are of $\frac{7}{16}$-inch thick plate. The lower flues are of $\frac{5}{16}$-inch thick plate. The tube-plates are of ½-inch thick iron. The tubes are lap-welded and the iron is 0·151 inch thick. All other parts of the boiler are of ⅜-inch thick plate.

The grate-bars are in two lengths; they are 1 inch wide on top, and have ⅝-inch wide air spaces between them.

The following are the principal dimensions and proportions, namely:—

| | |
|---|---|
| Length of boiler, | 22 feet. |
| Breadth of boiler, | 9 " |
| Height of boiler, exclusive of steam chimney, | 9 " 3 inches. |
| Height of steam chimney above top of shell of boiler, | 8 " |
| Diameter of steam chimney, | 7 " |
| Number of furnaces, | 3. |
| Breadth of furnaces, | 2 feet 4 inches. |
| Length of fire grates, | 6 " 9 " |
| Total area of grate surface, | 47¼ square feet. |
| Height from crown of furnace to bottom of ash-pit, | 3 feet 2½ inches. |
| Width of smoke connexion behind bridge wall, | 2 " |
| Number of lower flues, | 6. |
| Length of lower flues, | 9 feet 8 inches. |
| Diameter of lower flues, | two of 20 inches, two of 16 inches, and two of 15½ inches. |
| Width of smoke connexion at back end of boiler, | 2 feet. |
| Number of tubes, | 34. |
| Exterior diameter of tubes, | 6 inches. |
| Length of tubes, | 14 feet 11 inches. |
| Number of superheating vertical flues in steam chimney, | 4. |
| Diameter of superheating flues, | 17½ inches. |
| Length of superheating flues, | 8 feet. |
| Diameter of smoke-pipe, | 3 " |
| Height of smoke-pipe above grates, | 50 " |

| | |
|---|---|
| Width of water bottoms and water legs, | 6 inches. |
| Area of heating surface in the furnaces, | 120 square feet. |
| Area of heating surface in the smoke connexions behind bridge walls, | 100 " |
| Area of heating surface in the lower flues, | 315 " |
| Area of heating surface in the smoke connexion at back end of boiler, | 98 " |
| Area of heating surface in the smoke connexion at front end of boiler, | 91 " |
| Area of heating surface in the tubes (calculated for their interior circumference), | 760 " |
| Total area of water heating surface, | 1484 " |
| Aggregate cross area of lower flues, | 9·770 " |
| Aggregate cross area of tubes, | 6·131 " |
| Aggregate cross area of superheating flues, | 6·681 " |
| Aggregate cross area of the chimney, | 7·068 " |
| Ratio of heating to grate surface, | 31·408 to 1·000. |
| Ratio of grate surface to cross area of lower flues, | 4·836 " |
| Ratio of grate surface to cross area of tubes, | 7·707 " |
| Ratio of grate surface to cross area of superheating flues, | 7·072 " |
| Ratio of grate surface to cross area of the chimney, | 6·685 " |
| Distance traversed by the products of combustion from the centre of the furnaces to their delivery into the uptake, | 34 feet 6 inches. |
| Steam room in the boiler, exclusive of steam chimney, | 782 cubic feet. |
| Steam room in the steam chimney, | 265 " |
| Total steam room, | 1047 " |
| Weight of water contained in the boiler to six inches above tubes, | 34,100 pounds. |
| Weight of the boiler, including all doors and plates, but exclusive of other appendages, | 55,100 " |
| Area of steam superheating surface in the uptake and vertical flues, | 182 7 square feet. |

## RESULTS.

With the boiler described and the experiment conducted in the manner narrated, the following results were obtained, namely:—

| | |
|---|---|
| Date of commencing the experiment, | 1 P. M. May 15, 1862. |
| Date of ending the experiment, | 1 P. M. May 18, 1862. |
| State of the weather, | Clear with light breezes. |

| | | |
|---|---|---|
| TOTAL QUANTITIES. | Duration of the experiment in consecutive hours, | 72· |
| | Number of cubic feet of water evaporated, | 5,577·65 |
| | Number of pounds of water evaporated, | 347,710·70 |
| | Number of pounds of Blackheath anthracite consumed, | 38,724· |
| | Number of pounds of refuse from the anthracite in ashes, &c., | 5,517· |
| | Number of pounds of combustible consumed, | 33,207· |
| | Per centum of the anthracite in refuse, | 14·25 |

| | | |
|---|---|---|
| MEAN QUANTITIES. | Temperature in degrees Fahr. of the external atmosphere in the shade, | 76· |
| | Temperature in degrees Fahr. of the fire room, | 92·3 |
| | Temperature in degrees Fahr. of the water in the tank, | 59·2 |
| | Barometer, | 30·1 |
| | Thickness of the bed of anthracite upon the grates, in inches, | 8· |
| | Pounds of anthracite consumed per hour per square foot of grates, | 11·383 |
| | Pounds of combustible consumed per hour per square foot of grates, | 9·757 |
| EVAPORATION. | Total number of pounds of water evaporated from temperature of 100° Fahr., | 360,868·733 |
| | Total number of pounds of water evaporated from temperature of 212° Fahr., | 402,701·224 |
| | Pounds of water evaporated from temperature of 100° Fahr. by one pound of anthracite, | 9·319 |
| | Pounds of water evaporated from temperature of 212° Fahr. by one pound of anthracite, | 10·399 |
| | Pounds of water evaporated from temperature of 100° Fahr. by one pound of combustible, | 10·867 |
| | Pounds of water evaporated from temperature of 212° Fahr. by one pound of combustible, | 12·127 |

In the above table the weight of the cubic foot of water is taken at 62·34 pounds, the temperature being 59°·2 Fahr. The total heat of steam of atmospheric pressure with the barometer at 30·1 inches is taken at 1178°·17 Fahr. above zero.

What is termed "combustible" in the table is the remainder of the anthracite after subtraction of the refuse.

The economic results are given for what would have been the evaporation had the feed-water been supplied to the boiler at the temperature of 100° and 212° Fahr. instead of 59°·2, in order that they may be comparable with those from other boilers.

It will be observed that the evaporative efficiency of this boiler is very high. All the conditions are favorable for a maximum result. The boiler being new was perfectly clean; the combustion was regular and unforced; the calorimeter was smaller than the proportion to grate usually given in practice; the heating surface was unusually large in proportion to grates; there was a constant admission of air to the furnace above the fuel; the thickness of the bed of coal upon the grates was in good proportion to the size of the lumps; the lower flues afforded an ample mingling chamber for the gases, and, the smoke connexions being in common for all the furnaces, maintained a uniform temperature in the flues and tubes, notwithstanding the alternations of firing and cleaning in the different furnaces.

# EXPERIMENTS

## TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER

OF

## THE UNITED STATES STEAMER

# "MOUNT VERNON,"

WITH PENNSYLVANIA ANTHRACITE, AND WITH CUMBERLAND SEMI-BITUMINOUS COAL.

# EXPERIMENTS

## TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER

OF THE

## UNITED STATES STEAMER "MOUNT VERNON,"

WITH PENNSYLVANIA ANTHRACITE, AND WITH CUMBERLAND SEMI-BITUMINOUS COAL.

---

THE boiler of the U. S. Steamer "MOUNT VERNON," on which the following experiments were made, was entirely new, never having had steam on previously. It had just been tested and found tight with a hydrostatic pressure of 60 pounds per square inch above the atmosphere, placed in the vessel, and covered with new thick felt sewed on thick canvass. The top of the boiler to its vertical sides, and the entire steam chimney or drum, had, additionally, and over the felt, sheet lead soldered together.

The experiments were made at the dock in Baltimore. The evaporation was effected under the atmospheric pressure. The steam escaped through a temporary pipe of 12 inches diameter fitted to a hole of the same diameter near the top of the side of the steam chimney. The elbow of the pipe, after leaving the hole, declined a little and had its bottom pierced to allow the escape of whatever water might result from the condensation of steam in the pipe, so that it should not enter the boiler for re-evaporation.

The feed-water was accurately measured in an iron tank of 128·154 cubic feet capacity. This tank was situated on the dock above the level of the boiler, and was filled each time to its brim, through a hose and stop-cock by gravity, from the pipes supplying the city with water; it was entirely evacuated each time by gravity through a hose and stop-cock leading from its bottom to the boiler.

The coals used for the experiments were a Pennsylvania anthracite from the Susquehanna Valley; of good quality, in lumps of egg size and free from dust. And a similar quality of semi-bituminous coal from the Cumberland mines of Maryland.

The experiment with the anthracite was made first, and was followed by that with the semi-bituminous coal as soon after as the furnaces could be cleaned and re-wooded, and the tubes swept. Both experiments were conducted precisely alike, and in exactly the manner described for those made with the boiler of the U. S. Steamer "VALLEY CITY." They were made by the Engineer Department of the vessel with similar instruments, and a similar hourly record of the data was kept. Owing to the breaking of the high grade thermometer, and the impossibility of procuring another in time, the temperature of the products of combustion in the uptake during the experiment with the anthracite, was lost: it was taken, however, during the experiment with the semi-bituminous coal.

The following is the description of the boiler used.

## BOILER.

The boiler is of the type with horizontal fire-tubes returned above the furnaces.

The shell is flat on bottom, sides, and ends, and has a semi-cylindrical top. The length on the bottom, and up to the bottom of the uptake, is 15 feet 3 inches; from that point the front projects over until at the top the shell is 17 feet 6 inches long.

The furnaces are three in number, each 3 feet 6 inches wide in the clear with a fire-grate 7 feet 6 inches long. They are semi-cylindrical on top. Height from bottom of ash-pit to crown of furnace 3 feet 9 inches. The angles of the ash-pit are rounded on a radius of 12 inches. The water-bottom below the ash-pits is 6 inches wide from out to out of metal. The water-legs between and at the sides of the furnaces are 5 inches wide from out to out of metal. The water space at the front of the furnaces is also 5 inches wide.

The opening for the furnace door is semi-circular on top, extreme height 15 inches, breadth 18 inches. The door is of cast iron, perforated with fifteen holes of 1 inch diameter for the admission of air to the furnace over the fuel. The door has a box lining plate of wrought iron pierced with forty-one holes of $\frac{3}{4}$ inch diameter.

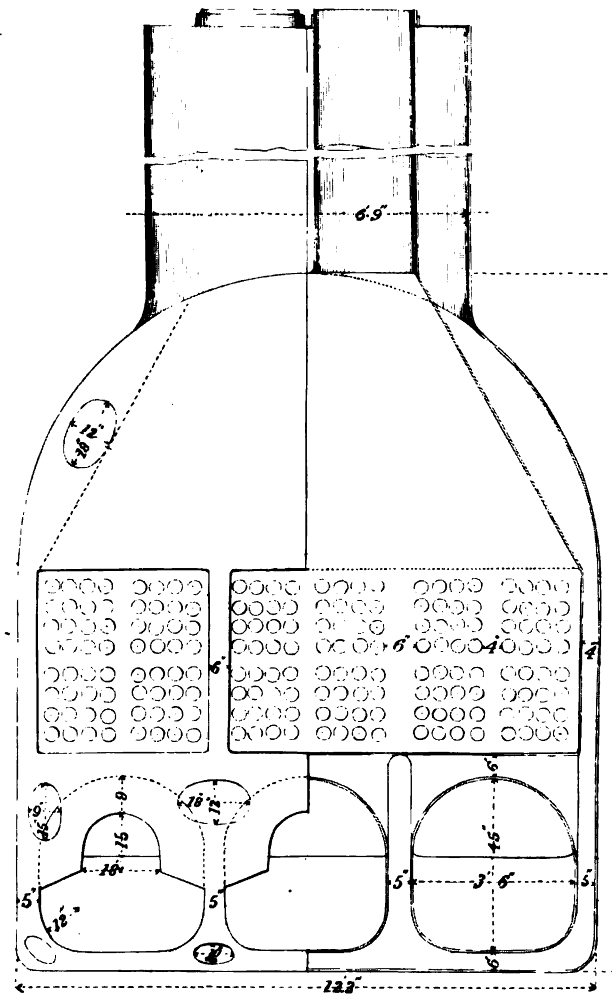
At the back of the furnace a combustion chamber is formed by a prolongation of the semi-cylindrical top of the furnace. The cross section of this chamber is a semi-circle of 3 feet 6 inches diameter; it is 4 feet 7 inches long, and extends to the back smoke connexion.

The back smoke connexion is in common for all three furnaces. It is a parallelopipedon in form, of 21 inches wide, 11 feet 6 inches broad, and 6 feet 6 inches high. At the bottom, water-legs 5 inches wide and 2 feet 3 inches high are carried up in continuation of the water-legs between the furnaces, and between the combustion chambers, to give the heated gases an upward direction.

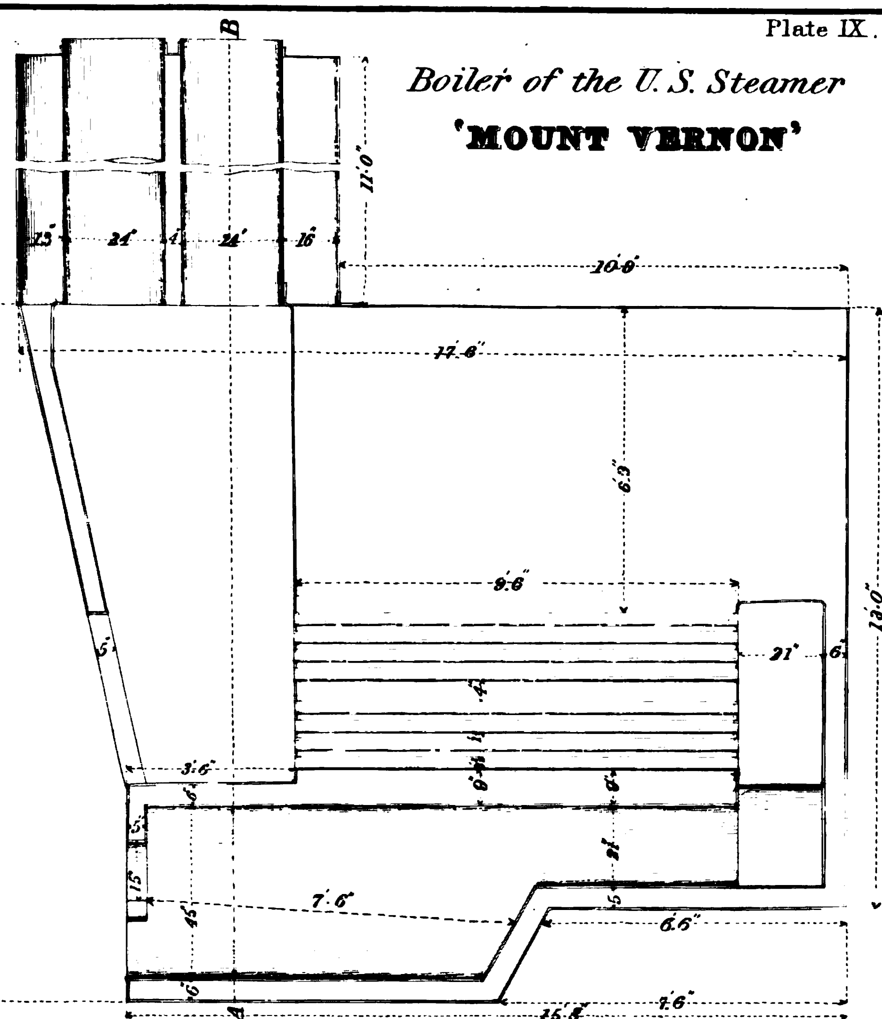
The tubes extend horizontally from the back smoke connexion to the uptake; they are two hundred and sixteen in number. They are $3\frac{1}{2}$ inches in external and $3\frac{1}{4}$ inches in internal diameter. Their extreme length is 9 feet 6 inches. They are arranged in nine rows vertically and twenty-four rows horizontally. The distance in the clear horizontally between them over the water-legs between the furnaces is 6 inches; over the centre of each furnace 4 inches; at other places $1\frac{1}{2}$ inch. The distance in the clear vertically between them between the fourth and fifth rows from the bottom is 4 inches; at other places $1\frac{1}{2}$ inch. The

Plate IX.

## Boiler of the U. S. Steamer 'MOUNT VERNON'




Half Elevation. Half Section on A.B.




Longitudinal Section through centre of shell, furnace, tubes, and uptake.

distance in the clear between the bottom of the bottom rows of tubes and the top of the furnaces and combustion chambers is 9 inches. The total height occupied by the tubes is 46 inches.

The lower part of the uptake, for a height of 4 feet 3 inches is rectangular; width on bottom 3 feet inside of water-legs, breadth 11 feet 6 inches; width at 4 feet 3 inches high, 4 feet. Distance from bottom of uptake to top of furnace, 6 inches. The total height of the uptake is 10 feet 3 inches, and at its top it is drawn into a square of 5 feet. From this square rise four superheating flues of 24 inches internal diameter and 11 feet length, separated by spaces 4 inches wide. These flues deliver the products of combustion into the smoke-pipe, which stands immediately over them and has its lower part enlarged to embrace them.

The superheating flues are surrounded by a steam chimney or drum 6 feet 9 inches in diameter and 11 feet in height above the top of the boiler shell. This system of carrying the products of combustion through circular flues in the steam chimney, avoids the use of bracing in the steam chimney, besides having the advantage of drying or superheating the steam. The gain is, therefore, both in the economy of construction, and in the use of the fuel.

At the front of the uptake, and immediately over the spaces between the furnaces, there are water-legs 6 inches square in section, and 4 feet 1 inch in height. The openings between these legs are closed by doors of cast iron fitting air-tight against cast iron frames bolted around the openings and having faced projections against which similar projections on the door close. The doors are lined with a wrought iron plate, distant 1 inch, which space is filled with a mixture of plaster of Paris and ashes.

Manholes are made in the steam room of the shell, and also in the spandrels of the furnaces. Handholes are made in the spandrels of the ash-pits at front and back. The water spaces at the bottom of the combustion chamber, and at the back of the after smoke connexion, are 6 inches wide.

The bottoms of the ash-pits, and of the shell of the boiler, are of $\frac{7}{16}$-inch thick plate. The tube plates are $\frac{5}{8}$-inch thick. All other parts are of $\frac{3}{8}$-inch thick plate. All the seams not in connexion with the fire are double riveted. The flat water spaces are braced by socket bolts every 9 inches. Other parts of the shell are braced every 12 inches by angle iron, and rods of $1\frac{1}{2}$ inch diameter.

The following are the principal dimensions and proportions, namely:—

| | |
|---|---|
| Extreme length of boiler on bottom, | 15 feet 3 inches. |
| Extreme length of boiler on top, | 17 " 6 " |
| Extreme breadth of boiler, | 12 " 2 " |
| Extreme height of boiler, exclusive of steam chimney, | 15 " |
| Extreme height of boiler, inclusive of steam chimney, | 26 " |
| Number of furnaces, | 3. |
| Length of each furnace, | 7 feet 6 inches. |
| Breadth of each furnace, | 3 " |
| Total area of grate surface, | 78·75 square feet. |
| Number of tubes, | 216. |
| External diameter of tubes, | $3\frac{1}{2}$ inches. |
| Internal diameter of tubes, | $3\frac{1}{4}$ " |
| Length of tubes, | 9 feet 6 inches. |

| | |
|---|---|
| Heating surface in the furnaces, | 174·45 square feet. |
| Heating surface in the combustion chambers, | 123·75 " |
| Heating surface in the back smoke connexion, | 192·70 " |
| Heating surface in the tubes, calculated for internal diameter, | 1,745·91 " |
| Heating surface in the uptake to 12 inches above tubes, | 137·19 " |
| Total area of water heating surface, | 2,374·00 " |
| Number of steam superheating flues, | 4. |
| Internal diameter of superheating flues, | 2 feet. |
| Length of superheating flues, | 11 " |
| Area of superheating surface in superheating flues, | 276·45 square feet. |
| Area of superheating surface in uptake, from 12 inches above tubes, | 136·55 " |
| Total area of steam superheating surface, | 413·00 " |
| Diameter of smoke-pipe, | 4 feet. |
| Height of smoke-pipe above grates, | 55 " |
| Aggregate cross area of combustion chambers for draught, | 14·350 |
| Aggregate cross area of tubes for draught, | 12·442 |
| Aggregate cross area of superheating flues for draught, | 12·564 |
| Cross area of the smoke chimney, | 12·566 |
| Distance traversed by the products of combustion from the centre of the furnace to their delivery into the uptake, | 23 feet 6 inches. |
| Capacity of steam room in the boiler, | 738 cubic feet. |
| Capacity of steam room in the steam chimney, | 267 " |
| Capacity of steam room in the boiler and steam chimney, | 1,005 " |
| Weight of water in the boiler up to 12 inches above top of tubes, | 54,600· pounds. |
| Weight of boiler, exclusive of smoke-pipe and grate-bars, but inclusive of all doors and plates, | 77,643· " |
| Weight of smoke-pipe, | 6,844· " |
| Weight of grate-bars, bearers, &c., | 4,916· " |
| Ratio of the water heating to the grate surface, | 30·146 to 1·000. |
| Ratio of the grate surface to the cross area of the combustion chambers, | 5·488 " |
| Ratio of the grate surface to the cross area of the tubes, | 6·329 " |
| Ratio of the grate surface to the cross area of the superheating flues, | 6·268 " |
| Ratio of the grate surface to the cross area of the smoke-pipe, | 6·267 " |
| Ratio of the steam superheating surface to the grate surface, | 5·244 " |

With the above described boiler, and the experiments conducted in the manner narrated, the following data and results were obtained, namely :—

TABLE CONTAINING THE DATA AND RESULTS OF THE EXPERIMENTS TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER OF THE U. S. STEAMER "MOUNT VERNON," WITH PENNSYLVANIA ANTHRACITE, AND WITH CUMBERLAND SEMI-BITUMINOUS COAL.

| | | ANTHRACITE. | SEMI-BITUMINOUS COAL. |
|---|---|---|---|
| | Date of commencing the experiment, | 11 A. M. Sept. 18, 1862. | 12·40 P. M. Sept. 20, 1862. |
| | State of the weather, | Clear with very light breeze. | Clear with light breeze. |
| TOTAL QUANTITIES. | Duration of the experiment, in consecutive hours, | 49 | 48⅓ |
| | Number of cubic feet of water evaporated, | 6,023·2 | 5,724·2 |
| | Number of pounds of water evaporated, | 374,856·948 | 356,382·550 |
| | Number of pounds of coal consumed, | 45,205· | 43,834· |
| | Number of pounds of refuse from the coal, in ashes, clinker, &c., | 4,528· | 6,608· |
| | Number of pounds of combustible consumed, | 40,677· | 37,226· |
| | Per centum of the coal in refuse, | 10·02 | 15·07 |
| MEAN QUANTITIES. | Temperature in degrees Fahr. of the external atmosphere in the shade, | 69·5 | 68·7 |
| | Temperature in degrees Fahr. of the fire room, | 91·9 | 86·0 |
| | Temperature in degrees Fahr. of the water in the tank, | 74·6 | 73·3 |
| | Temperature in degrees Fahr. of the products of combustion in the uptake, | . . | 259·6 |
| | Barometer, | 30·07 | 30·12 |
| | Thickness of the bed of coal upon the grates, in inches, | 8· | 8· |
| | Pounds of coal consumed per hour per square foot of grates, | 11·715 | 11·516 |
| | Pounds of combustible consumed per hour per square foot of grates, | 10·541 | 9·780 |
| EVAPORATION. | Total number of pounds of water evaporated from temperature of 100° Fahr., | 383,688·238 | 365,208·155 |
| | Total number of pounds of water evaporated from temperature of 212° Fahr., | 428,167·384 | 407,544·117 |
| | Pounds of water evaporated from temperature of 100° Fahr. by one pound of coal, | 8·488 | 8·332 |
| | Pounds of water evaporated from temperature of 212° Fahr. by one pound of coal, | 9·472 | 9·297 |
| | Pounds of water evaporated from temperature of 100° Fahr. by one pound of combustible, | 9·432 | 9·811 |
| | Pounds of water evaporated from temperature of 212° Fahr. by one pound of combustible, | 10·526 | 10·948 |

# EXPERIMENTS

## TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER

OF

## THE UNITED STATES STEAMER

# "VALLEY CITY,"

WITH PENNSYLVANIA ANTHRACITE, AND WITH CUMBERLAND SEMI-BITUMINOUS COAL.

# EXPERIMENTS

## TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER

OF THE

## UNITED STATES STEAMER "VALLEY CITY,"

### WITH PENNSYLVANIA ANTHRACITE, AND WITH CUMBERLAND SEMI-BITUMINOUS COAL.

---

The boiler of the U. S. Steamer "Valley City," on which these experiments were made, was entirely new, never having been used before. It had just been placed in the vessel, and was covered with thick felt sewed on canvass, over which a lead sheathing was soldered. Its perfect tightness had been assured under a hydrostatic pressure of 60 pounds per square inch above the atmosphere.

The experiments were made at the dock in Baltimore, and the evaporation was effected under the atmospheric pressure. The steam escaped through a pipe of 6 inches diameter fitted to a hole of the same area in the side of the steam chimney or drum near its top. The elbow of this pipe on leaving the steam chimney was a little drooped, and its bottom pierced so as to prevent the water that might result from the condensation of steam in the pipe from re-entering the boiler.

The feed-water was obtained from the pipes supplying the city, and, before entering the boiler, was carefully measured in an iron tank which was filled each time to precisely 57·188 cubic feet capacity. The water entered the tank by gravity through a hose and stop-cock, and was discharged into the boiler by gravity through another hose and stop-cock: a few minutes only were required for the filling of the tank.

The coals used for the experiments were of two kinds, namely:—a Pennsylvania anthracite from the Susquehanna Valley, of fair merchantable quality, in lumps of egg size, and free from dust. And a similar quality of semi-bituminous coal from the Cumberland mines of Maryland. The experiment with the anthracite was made first, and was almost immediately followed by that with the semi-bituminous coal, the

interval of one hour between them being employed in sweeping the tubes, &c., and in starting the second fires. Both experiments were conducted precisely alike, and in the following manner, namely:—

The boiler having been filled with water to 9 inches above the top of the tubes, a fire of wood was made in each furnace and the water brought by it to the boiling point. The wood was now burned down to just enough live embers to kindle the coal which was then fired, the time taken, the exact height of the water in the boiler noted on a glass gauge, and the experiment held to commence. Each experiment continued precisely 48 hours, and at its expiration the height of the water in the boiler was made exactly the same as at the commencement. The coal was allowed to burn out as nearly as possible at the end of the 48 hours, when the furnaces were drawn and the unconsumed coal separated from the ashes, weighed, and deducted from the total amount consumed. All the coal consumed during the experiment was carefully weighed and fired with great uniformity: the fires were carried 8 inches thick and kept clean and even. The combustion was not forced, but the furnaces were fed with all the coal they would properly consume. The chimney-damper and the holes in the furnace doors were all kept open.

The experiments were conducted by the Engineer Department of the vessel. Regular watches were kept and a tabular record in which were noted hourly the weight of coal thrown in the furnaces during that hour; the temperature of the external atmosphere in the shade; of the fire room; of the water in the tank; and of the products of combustion in the uptake. Also the height of the barometer. In appropriate columns were entered the time when each tankful of water was emptied, and the weight of ashes, &c., withdrawn from the furnaces. Particular care was taken to weigh the ashes in the *dry* state; in this weight, as composing the total refuse from the coal, were included the ashes, &c., from the furnaces at the end of the experiment, and the sweepings of the tubes. The fires were cleaned at regular intervals. The state of the weather was noted on the record by each Engineer during his watch.

The following is the description of the boiler used:—

## BOILER.

The boiler has horizontal flues in direct continuation of the furnaces, with horizontal tubes returned above the flues and furnaces.

The shell consists of a front rectangular in plan and semi-cylindrical on top, which contains the two furnaces. Width of the front 8 feet; length of the front 6 feet 10 inches; height of the front, exclusive of steam chimney or drum, 9 feet 6 inches. And of a cylindrical part behind the front of 8 feet diameter and 7 feet 2 inches length. The upper semi-circumference of this cylindrical part is a continuation of the semi-cylindrical top of the front, which leaves a height of (9 feet 6 inches less 8 feet =) 1 foot 6 inches between the bottom of the front and the bottom of the cylindrical part.

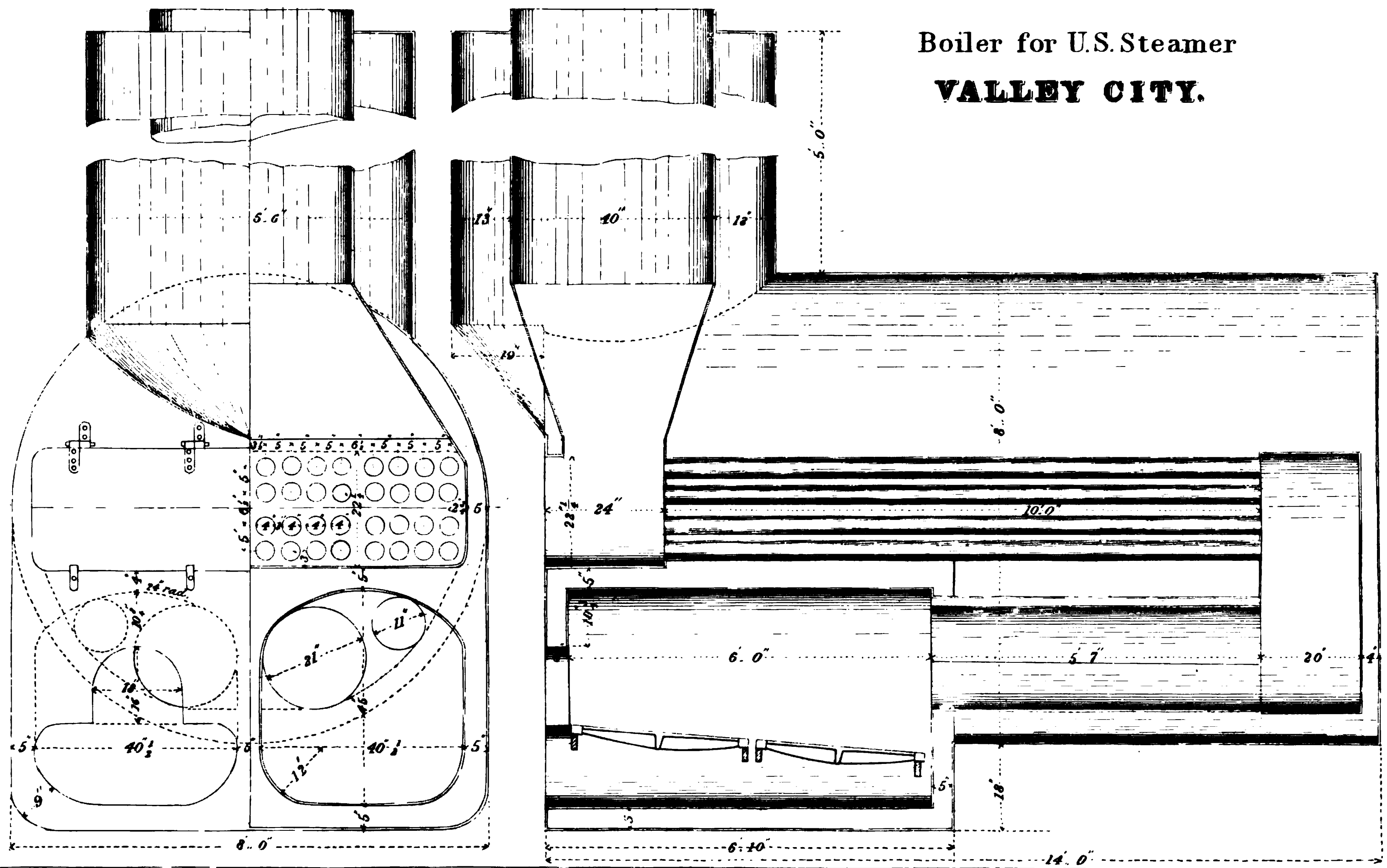
The furnaces are two in number, each 40 inches in width and 6 feet in length; the crown is a curve struck from three centres with radii of 10½ and 24 inches. The corners of the ash-pits are rounded with a radius of 12 inches. Height from bottom of ash-pit to crown of furnace 45 inches. The furnaces are separated from each other by a water-leg of 6 inches width, and from the shell of the boiler by a water-leg of 5 inches width. The width of the water-bottom beneath ash-pit is 5 inches. The water spaces at front and back of furnaces are also 5 inches wide.

The opening for the furnace door is semi-circular on top; it is 18 inches wide and 16 inches high. The door is perforated with eleven holes of 1 inch diameter, and is provided with a box lining perforated with

Plate X.

Boiler for U.S. Steamer

**VALLEY CITY.**



forty-six holes of ½-inch diameter, for the admission of air to the furnaces above the fuel. Area of air opening in each furnace door 8·6394 square inches; in the lining plate of the door 9·0298 square inches.

From the back of each furnace, there go direct (without the intervention of any smoke connexion) two cylindrical horizontal flues of 5 feet 7 inches length; inner diameter of one 21 inches, of the other 11 inches.

The flues from both furnaces discharge into one back smoke connexion 20 inches wide and 4 feet 4¾ inches extreme height. The top of this connexion is flat, and the bottom and sides are concentric with the cylindrical part of the shell of the boiler, from which it is separated by a water space 5 inches wide. A water space of 4 inches width separates it from the flat back of the boiler.

From the upper part of the smoke connexion there are returned over the flues and furnaces, and towards the front of the boiler, sixty-four horizontal tubes of 4 inches external diameter and 10 feet length. The tubes are in four rows vertically and sixteen rows horizontally. The space horizontally between them at the centre of the boiler is 3 inches in the clear; at the centre of each furnace 2½ inches in the clear; at other places 1 inch in the clear. The second and third rows of tubes, vertically, are separated by a space of 2½ inches in the clear; at other places 1 inch in the clear. The space between the top of the furnace and the bottom of the lowest row of tubes is 6 inches in the clear. Total height occupied by tubes 20½ inches.

The tubes discharge into one rectangular uptake of 20 inches breadth, 7 feet 2 inches width, and 22½ inches height; the upper part of which in a height of 34½ inches is drawn into a circle of 40 inches diameter for the smoke-pipe. This circular part extends through the steam chimney or drum for a further height of 5 feet 2½ inches.

The uptake is fitted with two cast iron doors, lined with a wrought iron plate, and having the intervening space of 1 inch filled with a mixture of plaster of Paris and ashes.

The upper cylindrical part of the uptake is surrounded by a steam chimney of 5 feet 6 inches diameter, concentric with it, and 5 feet high above the top of the boiler shell. This steam chimney projects beyond the front of the boiler 19 inches.

The shell of the boiler is double riveted and braced in all its flat surfaces every 9 inches. The steam chimney is also braced to the cylindrical part of the uptake. Manholes are placed in the spandrels of the furnaces and in the steam room of the boiler. Hand-holes are also placed in the spandrels of the ash-pits and at the back of the front and of the cylindrical part of the shell.

The thickness of the plate used for the bottom of the boiler front and for the bottom of the ash-pits is $\frac{7}{16}$-inch. The tube plates are ½-inch thick, and the tubes are expanded on one side of them and riveted over on the other. The 21 inches diameter flues are of $\frac{5}{16}$-inch thick plate; the 11 inches diameter flues are of ¼-inch thick plate; all other parts are of ⅜-inch thick plate.

The following are the principal dimensions and proportions of the boiler, namely:—

| | |
|---|---|
| Extreme breadth, | 8 feet. |
| Extreme length, exclusive of steam chimney, | 14 " |
| Extreme length, inclusive of steam chimney, | 15 " 7 inches. |
| Extreme height, exclusive of steam chimney, | 9 " 6 " |
| Extreme height, inclusive of steam chimney, | 14 " 6 " |

| | |
|---|---|
| Number of furnaces, | 2. |
| Length of each furnace, | 6 feet. |
| Breadth of each furnace, | 3 " 4 inches. |
| Total area of grate surface, | 40 square feet. |
| Number of tubes, | 64. |
| External diameter of tubes, | 4 inches. |
| Internal diameter of tubes, | 3¾ " |
| Length of tubes, | 10 feet. |
| Number of flues, | 4. |
| Internal diameter of flues, | two of 21 and two of 11 inches. |
| Length of flues, | 5 feet 7 inches. |
| Heating surface in the furnaces, | 114·00 square feet. |
| Heating surface in the flues, calculated for inside surface, | 93·54 " |
| Heating surface in the back smoke connexion, | 73·40 " |
| Heating surface in the tubes, calculated for inside surface, | 628·27 " |
| Heating surface in the uptake to 12 inches above top of tubes, | 44·50 " |
| Total area of water heating surface, | 953·71 " |
| Total area of steam heating surface in uptake above 12 ins. above top of tubes, | 180·00 " |
| Diameter of smoke-pipe, | 3 feet 4 inches. |
| Height of smoke-pipe above grates, | 34 " |
| Aggregate cross area of flues for draught, | 6·131 square feet. |
| Aggregate cross area of tubes for draught, | 4·909 " |
| Cross area of smoke-pipe for draught, | 8·726 " |
| Distance traversed by the products of combustion from the centre of the furnace to their delivery into the uptake, | 22 feet. |
| Capacity of steam room in the boiler, | 116 cubic feet. |
| Capacity of steam room in the steam chimney, | 75 " |
| Capacity of steam room in the boiler and steam chimney, | 191 " |
| Weight of water in the boiler up to 12 inches above top of tubes, | 24,200 pounds. |
| Weight of boiler, exclusive of smoke-pipe and grate-bars, but inclusive of all doors and plates, | 31,097 " |
| Weight of smoke-pipe, | 5,034 " |
| Weight of grate-bars, bearers, &c., | 2,531 " |
| Ratio of the water heating to the grate surface, | 23·843 to 1·000. |
| Ratio of the grate surface to the cross area of the flues, | 6·524 " |
| Ratio of the grate surface to the cross area of the tubes, | 8·148 " |
| Ratio of the grate surface to the cross area of the smoke-pipe, | 4·584 " |
| Ratio of the steam superheating surface to the grate surface, | 4·500 " |

With the boiler described and the experiments conducted in the manner related, the following results were obtained:—

TABLE CONTAINING THE DATA AND RESULTS OF THE EXPERIMENTS TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER OF THE U. S. STEAMER "VALLEY CITY," WITH PENNSYLVANIA ANTHRACITE, AND WITH CUMBERLAND SEMI-BITUMINOUS COAL.

| | | ANTHRACITE. | SEMI-BITUMINOUS COAL. |
|---|---|---|---|
| | Date of commencing the experiment, | Meridian, Sept. 1, 1862. | 1 P. M. Sept. 3, 1862. |
| | State of the weather, | First 24 hours, cloudy with damp air; no breeze and occasional showers of rain. Last 24 hours clear and cool with light breezes. | Clear and cool with light breezes. |
| TOTAL QUANTITIES. | Duration of the experiment in consecutive hours, | 48 | 48 |
| | Number of cubic feet of water evaporated, | 2,802·212 | 3,259·716 |
| | Number of pounds of water evaporated, | 174,313·633 | 202,778·002 |
| | Number of pounds of coal consumed, | 21,200· | 22,990· |
| | Number of pounds of refuse from the coal, in ashes, clinker, &c., | 3,600· | 2,870· |
| | Number of pounds of combustible consumed, | 17,600· | 20,120· |
| | Per centum of the coal in refuse, | 17·00 | 12·48 |
| MEAN QUANTITIES. | Temperature in degrees Fahr. of the external atmosphere in the shade, | 70·2 | 69·6 |
| | Temperature in degrees Fahr. of the fire room, | 95·1 | 101·9 |
| | Temperature in degrees Fahr. of the water in the tank, | 78·0 | 78·0 |
| | Temperature in degrees Fahr. of the products of combustion in the uptake, | 284·4 | 345·2 |
| | Barometer, | 29·93 | 30·19 |
| | Thickness of the bed of coal upon the grates, in inches, | 8· | 8· |
| | Pounds of coal consumed per hour per square foot of grates, | 11·041 | 11·974 |
| | Pounds of combustible consumed per hour per square foot of grates, | 9·167 | 10·480 |
| EVAPORATION. | Total number of pounds of water evaporated from temperature of 100° Fahr., | 177,870·829 | 206,910·459 |
| | Total number of pounds of water evaporated from temperature of 212° Fahr., | 198,492·039 | 230,895·112 |
| | Pounds of water evaporated from temperature of 100° Fahr. by one pound of coal, | 8·390 | 9·000 |
| | Pounds of water evaporated from temperature of 212° Fahr. by one pound of coal, | 9·363 | 10·048 |
| | Pounds of water evaporated from temperature of 100° Fahr. by one pound of combustible, | 10·106 | 10·284 |
| | Pounds of water evaporated from temperature of 212° Fahr. by one pound of combustible, | 11·278 | 11·475 |

# EXPERIMENT

## TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER

OF

## THE UNITED STATES STEAMER

# "CRUSADER,"

WITH BLACKHEATH ANTHRACITE.

# EXPERIMENT

## TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER

OF THE

## UNITED STATES STEAMER "CRUSADER,"

WITH BLACKHEATH ANTHRACITE.

---

THE following experiment was made upon the boiler of the United States Steamer "CRUSADER" to determine its evaporative efficiency with Blackheath anthracite. During the experiment the vessel lay alongside the wharf of the New York Navy Yard.

The boiler was new and had never been used before. Previous to placing it in the vessel it had been tested and found perfectly tight under a hydrostatic pressure of 40 pounds per square inch. When placed in the vessel it was entirely covered with one thickness of No. 1 felt stitched to No. 1 canvass, in addition to which the top of the shell to the vertical sides, and the whole of the steam drum and steam chimney, were covered with sheet lead.

The evaporation was performed under the atmospheric pressure; and to give the steam such free egress at a low velocity as would secure it from entraining particles of solid water, an escape-pipe was fitted to the hole for the main steam-pipe near the top of the steam chimney. This hole had an area of 77 square inches. The pipe was of equal area, and arranged with an elbow having a perforation at its lowest point to discharge the water condensed in the pipe and prevent its return to the boiler.

The weight of water evaporated was ascertained by measurement in a tank previous to being run into the boiler. The tank was situated on the wharf, above the top of the boiler, and was filled from the Navy Yard reservoir through a hose provided with a stop-cock; the filling required but a few minutes as the water in the reservoir had a considerable elevation over the tank. The boiler was filled from the tank through a hose delivering into the safety-valve chamber. The tank was of wood lined with sheet lead,

and was filled each time to precisely 54 cubic feet capacity. A glass water gauge attached to the boiler showed the exact level of the water within.

The boiler being filled to about 9 inches above the top of the tubes, a fire of wood was made in the furnaces, and the water brought to the boiling point. As soon as the wood was burned down to the few coals required for kindling the anthracite, the time and the exact height of water in the glass gauge were noted, the anthracite fired and the experiment commenced. During its continuance equal quantities of coal were weighed out hourly, and fired uniformly by experienced firemen under the supervision of three Assistant Engineers of the Navy who stood regular watches of four hours throughout the ninety-six hours it continued. At the close of the experiment the fires were burned out as nearly as possible and the furnaces drawn at the moment the time expired; the unconsumed coal was then carefully picked out, weighed, and deducted. The water at the end of the experiment was brought to precisely the same level as at its commencement.

The anthracite, though not screened, was free from dust, and was in lumps of uniform and proper size. Every pound was carefully weighed, as was also,—and in the dry state,—the ashes and other refuse from it. The bed of coal upon the grates was kept level and at the same thickness, and the fires were cleaned at regular intervals. The water level in the boiler varied but slightly, and every precaution was taken to secure uniformity in all the conditions throughout, and a maximum result.

During the progress of the experiment, which was continued for 96 consecutive hours, a tabular record was kept in which was entered by the watch Engineer at the end of each hour, the weight of coal fired, the height of the barometer, and the temperature of the external atmosphere, of the fire-room, of the water in the tank, and of the products of combustion as they emerged from the tubes into the chimney uptake. In other columns were noted the ashes, and the time at which each tankful of water was emptied. The totals and means of these quantities will be found in the table hereinafter given of the results of the experiment.

To obtain the temperature of the products of combustion, a copper pot filled with oil in which a thermometer was immersed, was suspended in the uptake immediately in front of the centre of the tubes and as close to them as possible without touching. The pot could be taken out through a small door made in the large smoke box door for that purpose and the thermometer read without sensibly falling.

The dampers were never closed, and the rate of combustion was the maximum that could be obtained. In each furnace door there were ten holes of 1 inch diameter, making an aggregate area of 7·854 square inches; and the lining of each door was perforated with ninety-one holes of $\frac{7}{16}$-inch diameter, making an aggregate area of 13·677 square inches. These holes were all open during the entire experiment.

## BOILER.

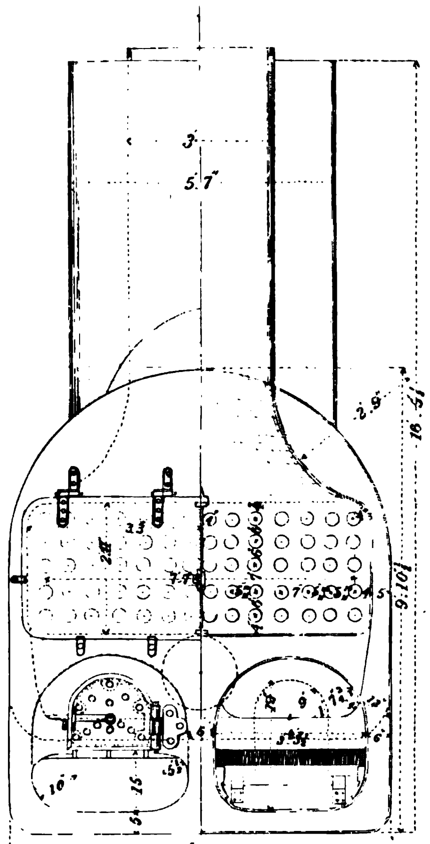
The boiler is of the horizontal tubular kind with the tubes returned above the furnaces.

The shell (excluding steam drum and steam chimney) is 16 feet 2 inches long, 8 feet 1 inch wide, and 9 feet 10½ inches high at front for the length of the furnaces, and 8 feet 11 inches high for the remainder of the length. The top is semicircular, and bottom and sides are flat.

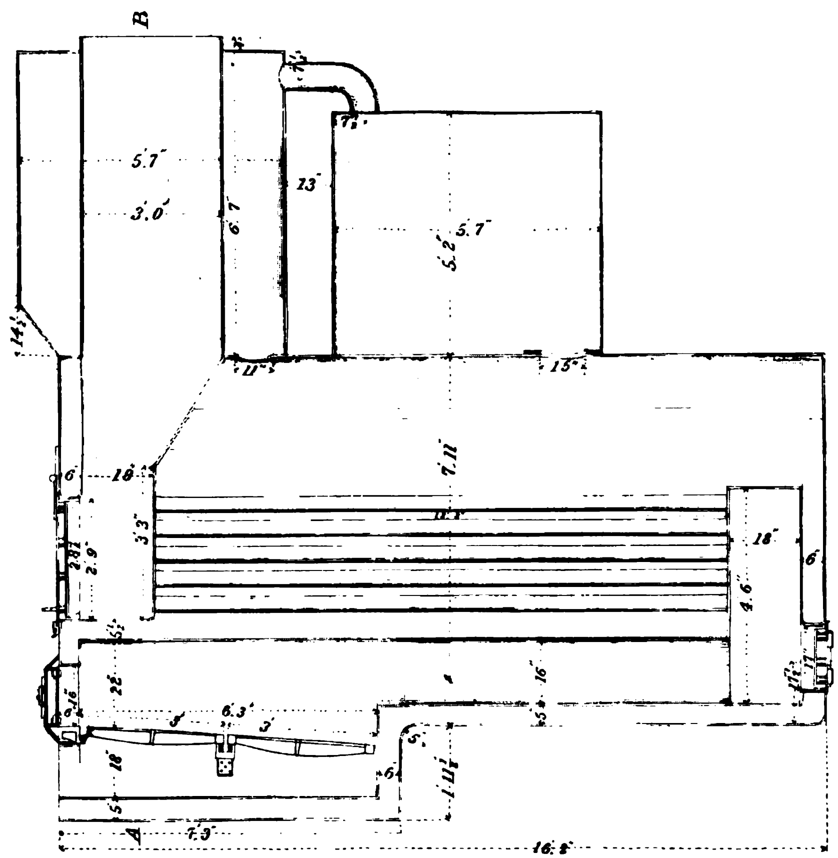
The smoke-pipe, which is 3 feet in diameter, is surrounded with a steam chimney 5 feet 7 inches in diameter, and 6 feet 7 inches in height above the top of the shell. The steam drum is immediately behind the steam chimney, and is 5 feet 7 inches in diameter, with a height of 5 feet 2 inches above the top of the shell. It is joined by a pipe to the steam chimney, and from the top of the latter the steam-pipe is carried to the engines.

Plate XI.

# BOILER OF U.S.S. 'CRUSADER'.



Half Elevation. Half Section on A.B.



Longitudinal Section through centre of Shell and Furnace.

The furnaces are two in number and semicircular on top. They are 6 feet 3 inches long, and 3 feet $3\frac{1}{2}$ inches wide. Their crown is 22 inches above the grate bars at front and 27 inches above them at the back.

Each furnace has a cast iron door pierced with ten holes of 1 inch diameter. The door lining of $\frac{3}{8}$-inch plate is perforated with ninety-one holes of $\frac{7}{16}$-inch diameter. The door fitted the shell extremely close. The furnace door opening is semicircular on top, extreme height 16 inches, width 18 inches.

The total height from the bottom of the ash-pits to the crown of the furnaces is 40 inches, leaving the height from bottom of ash-pit to top of grate-bars 18 inches at the front and 13 inches at the back. The angles of the ash-pits are rounded on a radius of 10 inches.

From each furnace a combustion chamber 7 feet 5 inches long extends to the back smoke connexion. This chamber is, on top, a continuation of the crown of the furnace; on the bottom it is flat; in cross section it is a segment, 16 inches high, of a circle of 3 feet $3\frac{1}{2}$ inches diameter. At the junction of the arc and chord the angle is rounded on a radius of 4 inches.

The back smoke connexion is in common for the two furnaces. It is 18 inches wide in the clear, 4 feet 6 inches high, and extends across the boiler.

The boiler contains 70 tubes. They are of iron 4 inches in external diameter, $3\frac{3}{4}$ inches in internal diameter, and 12 feet 2 inches in extreme length. The least space between them horizontally in the clear is 2 inches, and vertically the same. They are secured in their plates by being expanded on one side and riveted over on the other. There are five rows in height, and occupy vertically a space of 29 inches. The distance in the clear between the crown of the furnaces and combustion chambers, and the lowest tube, is $7\frac{1}{2}$ inches.

The front smoke connexion or uptake is 18 inches wide and extends across the boiler.

All flat water spaces are 6 inches wide with the exception of the ash-pit water bottom which is 5 inches wide.

The grate-bars are in two lengths; they are 1 inch wide on top and have $\frac{1}{2}$-inch wide air spaces between them.

A manhole is placed in each spandrel of the furnace arches, and a handhole is placed in each spandrel at the ash-pit corners. A door is placed in the back smoke connexion and the opening temporarily bricked up. The smoke box doors are lined and accurately fitted to the shell.

The bottoms of the ash-pits and of the front of the boiler are of $\frac{7}{16}$-inch thick plate; the tube plates are $\frac{1}{2}$-inch thick; the tubes are $\frac{1}{8}$-inch thick; all other parts of the boiler are of $\frac{3}{8}$-inch thick plate. All seams not in contact with the fire are double riveted.

The following are the remaining dimensions and proportions required to be known, namely:—

| | | |
|---|---|---|
| Area of grate surface in both furnaces, | 41·25 | square feet. |
| Area of heating surface in furnaces, | 39·062 | " |
| Area of heating surface in combustion chambers, | 50·750 | " |
| Area of heating surface in back smoke connexion, | 58·500 | " |
| Area of heating surface in tubes (calculated for their inside diameter), | 835·530 | " |
| Area of heating surface in uptake to 9 inches above top of tubes, | 72·820 | " |
| Total area of water heating surface, | 1056·662 | " |
| Total area of steam heating surface in uptake and steam chimney, | 86 100 | " |
| Area of water level, | 111·000 | " |
| Capacity of steam room in shell, steam chimney, and steam drum, | 403·050 | cubic feet. |
| Capacity of water space up to 9 inches above tubes, | 467·000 | " |
| Height of smoke-pipe above grate-bars, | 43 feet 6 inches. | |

| | |
|---|---|
| Cross area of smoke-pipe, | 7·068 square feet. |
| Cross area of the two combustion chambers, | 5·071 " |
| Cross area of the seventy tubes (calculated for their inside diameter), | 5·368 " |
| Ratio of the water heating to the grate surface, | 25·616 to 1·000. |
| Ratio of the grate surface to the cross area of the combustion chambers, | 8·134 " |
| Ratio of the grate surface to the cross area of the tubes, | 7·684 " |
| Ratio of the grate surface to the cross area of the smoke-pipe, | 5·836 " |
| Distance traversed by the products of combustion from the centre of the furnace to their emergence from the tubes, | 26 feet 6 inches. |
| Weight of boiler, including all doors, handhole and manhole plates, but exclusive of grates and chimney, | 50,117 pounds. |

## RESULTS.

With the boiler just described and the experiment conducted in the manner narrated, the following evaporative results were obtained, namely:—

| | |
|---|---|
| Date of commencing the experiment, | 3 P. M. Dec. 7, 1861. |
| Date of ending the experiment, | 3 P. M. Dec. 11, 1861. |
| State of the weather, | Light breezes and clear sky for half the time: the other half light breezes and foggy. |

| | | |
|---|---|---|
| TOTAL QUANTITIES. | Duration of the experiment in consecutive hours, | 96· |
| | Number of cubic feet of water evaporated, | 6,010·533 |
| | Number of pounds of water evaporated, | 374,993· |
| | Number of pounds of Blackheath anthracite consumed, | 42,545· |
| | Number of pounds of refuse from the anthracite in ashes, &c., | 5,348· |
| | Number of pounds of combustible consumed, | 37,197· |
| | Per centum of the anthracite in refuse, | 8·74 |
| MEAN QUANTITIES. | Temperature in degrees Fahr. of the external atmosphere in the shade, | 49·4 |
| | Temperature in degrees Fahr. of the fire room, | 59·1 |
| | Temperature in degrees Fahr. of the water in the tank, | 44·2 |
| | Temperature in degress Fahr. of the products of combustion entering the chimney, | 513·6 |
| | Barometer, | 30·12 |
| | Thickness of the bed of anthracite upon the grates, in inches, | 8·5 |
| | Pounds of anthracite consumed per hour per square foot of grate surface, | 10·744 |
| | Pounds of combustible consumed per hour per square foot of grate surface, | 9·393 |
| EVAPORATION. | Total number of pounds of water evaporated from temperature of 100° Fahr, | 394,400·883 |
| | Total number of pounds of water evaporated from temperature of 212° Fahr, | 440,121·422 |
| | Pounds of water evaporated from temperature of 100° Fahr. by one pound of anthracite, | 9·270 |
| | Pounds of water evaporated from temperature of 212° Fahr. by one pound of anthracite, | 10·345 |
| | Pounds of water evaporated from temperature of 100° Fahr. by one pound of combustible, | 10·603 |
| | Pounds of water evaporated from temperature of 212° Fahr. by one pound of combustible, | 11·832 |

It will be observed that the boiler was proportioned to give a maximum evaporative result. The tubes, comparatively to their diameter of 4 inches, were very wide apart (2 inches in the clear), and the flat water-legs and bottoms had also the large width of 6 inches. The furnace crown was semicircular, and the bottom of the lowest tube was 7½ inches in the clear above it. The uptake was narrow (18 inches). All of which were extremely favorable to the easy and rapid circulation of the water. In addition, too, the boiler was quite new and clean.

Between the furnace and back smoke connexion there intervened a combustion chamber of 7 feet 5 inches length, in which ample time was afforded for a complete intermingling of the furnace gases before the temperature had become too much reduced for the proper action of the chemical affinities. This connexion, too, being in common to both furnaces, was a great advantage in their alternate firing. Air was also admitted through perforations in the furnace doors. The calorimeter of the tubes, though too small for giving the maximum rate of combustion, was the most favorable for high evaporative effect, its proportion being a little over one-eighth of the grate surface. The rate of combustion was moderate, being only 10·744 pounds of coal per square foot of grates per hour, and the draught was proportionally slow.

The coal was of the first quality anthracite, and yielded only 8·74 per centum of refuse. It was fired with the utmost care and regularity.

All the steam in its escape had to pass over the steam heating surfaces in the uptake and steam chimney, by which any solid particles of water entrained by the current were doubtless evaporated. The height of 8 feet 3 inches above the water level at which the steam entered the escape pipe, was also favorable to dryness by giving time for the solid particles of water, if any were combined, to separate by their gravity. Under such favorable conditions the evaporation reached 9·270 pounds of water per pound of coal, or 10·603 pounds of water per pound of combustible from a temperature of 100° Fahr.

Notwithstanding these favorable conditions, it will be observed that the temperature of the products of combustion on emerging from the tubes was 513·6° Fahr., or 301° Fahr. above the temperature of the water and steam within the boiler although the heat absorbing surface in the water was 25·616 times the grate surface. Had the evaporation been conducted under a higher steam pressure, the temperature of the products of combustion entering the uptake would, of course, have been considerably higher.

In the preceding table of results what is termed "combustible" is the remainder of the coal after deducting the refuse. In calculating what would have been the evaporation had the temperature of the feed-water been 100° and 212° Fahr., instead of 44·2° Fahr., the total heat of steam with the barometer at 30·12 inches, is taken at 1178·15° Fahr.

# EXPERIMENTS

MADE BY ORDER OF THE U. S. NAVY DEPARTMENT,

ON THE

# VERTICAL WATER TUBE BOILER OF THE U. S. STEAMER

# "WYANDOTTE."

# EXPERIMENTS

MADE BY ORDER OF THE U. S. NAVY DEPARTMENT

ON THE

# VERTICAL WATER TUBE BOILER OF THE U. S. STEAMER

# "WYANDOTTE,"

HAVING ITS TUBES OVER THE FURNACES TO DETERMINE THE RELATIVE ECONOMIC EFFICIENCY BY THE REGULAR ARRANGEMENT OF FURNACE ADOPTED IN THE U. S. NAVAL SERVICE FOR BOILERS OF THIS TYPE, AND BY THE SAME WITH THE AMORY BRIDGE APPLIED.

ALSO,

TO DETERMINE THE RELATIVE ECONOMIC EVAPORATION BY BLACKHEATH ANTHRACITE AND BY BROAD TOP SEMI-BITUMINOUS COAL UNDER VARIOUS CONDITIONS OF THICKNESS OF FIRE AND OF ADMISSION AND SUPPRESSION OF AIR THROUGH HOLES IN THE FURNACE DOORS.

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THE following experiments were made by order of the Navy Department primarily to determine the economic value of Mr. JONATHAN AMORY's metallic curved combustion chamber for boiler furnaces, the object of which is to effect the better combustion of the fuel and to preserve the metal of the chamber. This combustion chamber, according to AMORY's patent, dated April 19th, 1859, is to be constructed of plate iron and placed immediately behind the bridge-wall. Its section in said patent, lengthwise the furnace, resembles a vertical horse shoe with the open part horizontal and uppermost, and is composed of an inner and outer plate enclosing a space between them which is in communication with the atmosphere. To this space air is to be copiously supplied from the exterior of the boiler by means of a tube, and thence delivered into the chamber through small holes in the inner plate on the curve concave towards the furnace, and through a narrow orifice between the outer and inner plate at the top of the chamber at the extremity of the curve convex towards the furnace. The section as described extends across the furnace from side to side.

It is assumed by the Patentee that the current of gases from the furnace in their passage over the open top of this chamber will descend from their straight course and rotate or eddy in the chamber, becoming therein mixed with the atmospheric air supplied as described, and which, it is also assumed, has not been furnished in the furnace in sufficient quantity to fully oxidize the carbon and hydrogen of the fuel. Further, the continuous passage of a large body of air between the plates is imperatively required for their preservation by carrying off the heat, and the temperature thus imparted to the air entering the combustion chamber is also assumed by the Patentee to be a meritorious part of his system. The "Claim" in the Letters Patent is for the combustion chamber placed, formed, and constructed substantially as described, and for the purposes stated.

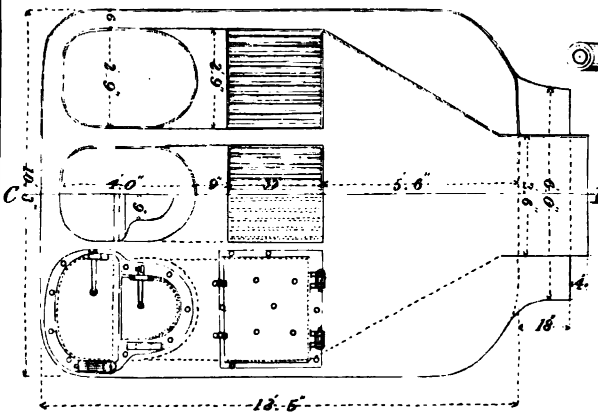
The Board ordered by the Navy Department to test and report upon this invention was composed of the Engineer-in-chief, B. F. Isherwood, and Commander C. H. Davis, U. S. N., and Professors Joseph Henry and A. D. Bache; but as they, owing to the pressure of other duties, could not give their personal attention to the necessary experiments, Chief Engineer B. F. Garvin was directed to make them at the New York Navy Yard with the aid of Assistant Engineers, Barlow, Musgrave, and Sprague, and to furnish the Board with the notes.

The boiler (Plate 12,) selected for the experiments had vertical water-tubes above the furnaces according to Martin's Patent, and was proportioned according to the usage in the Naval Service. It had just been built at the New York Navy Yard for the U. S. Steamer "Wyandotte" and had never been used. The shell was double riveted and, with the exception of the bottom and the bottoms of the ash-pits, which were of $\frac{7}{16}$-inch thick plate, was of $\frac{3}{8}$-inch thick plate. All other parts, except the tubes and tube-plates, were composed of $\frac{3}{8}$-inch thick plate. The tubes were of $\frac{1}{10}$-inch thick plate, lap-welded, and were placed with their centres in straight lines instead of being zigzagged as is usually done; the corner tubes over each furnace were omitted. The tube plates were $\frac{1}{2}$-inch thick. The crowns of the furnaces were semicircular, and the angles of the ash-pits were rounded on a radius of 12 inches.

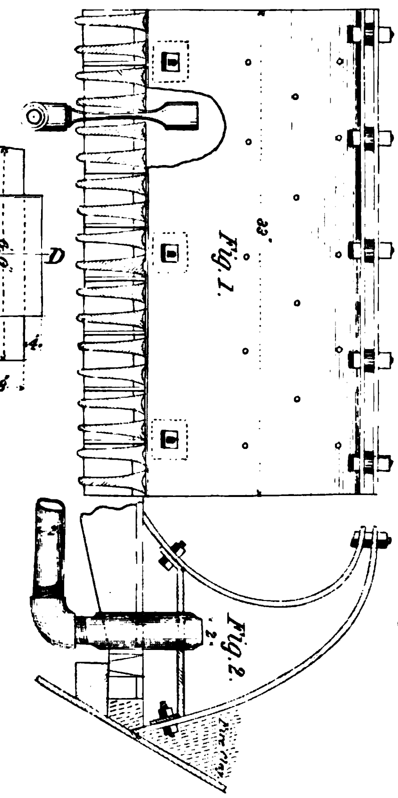
The following are the principal dimensions of the boiler, namely:—

| | |
|---|---|
| Length of boiler at level of furnaces, | 12 feet 2 inches. |
| Length of boiler on top, | 13 " 2 " |
| Width of boiler, | 10 " 3 " |
| Height of boiler, exclusive of steam chimney, | 13 " 5 " |
| Height of boiler, inclusive of steam chimney, | 14 " 11 " |
| Diameter of steam chimney or drum surrounding smoke-pipe, | 6 " |
| Diameter of smoke-pipe, | 3 " 6 " |
| Height of smoke-pipe above level of grates, | 50 " 9 " |
| Number of furnaces, | 3. |
| Height from bottom of ash-pit to crown of furnace, | 4 feet. |
| Width of furnace, | 2 " 9 inches. |
| Length of grate-bars, | 6 " 6 " |
| Height from centre of grate-bars to crown of furnace, | 2 " 3 " |
| Height between crown of furnace and bottom tube-plate at front of boiler, | 9 " |
| Height between crown of furnace and bottom tube-plate at back of boiler, | 6 " |
| Width of back smoke connexion, | 1 foot 6 " |
| Width of all water-bottoms and legs, | 6 " |

*Section on E.F.*




*Elevation.*

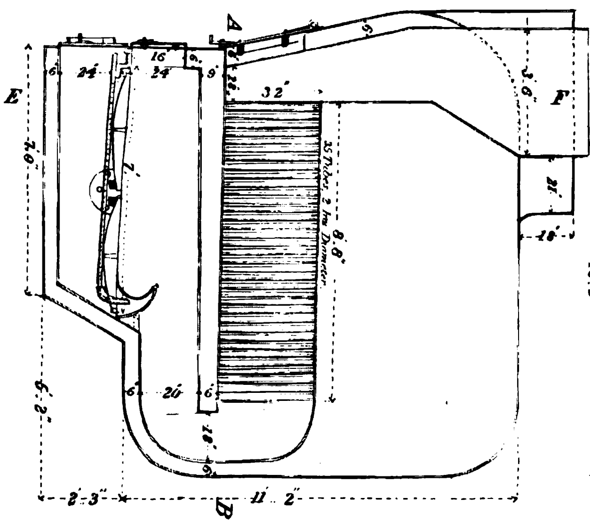


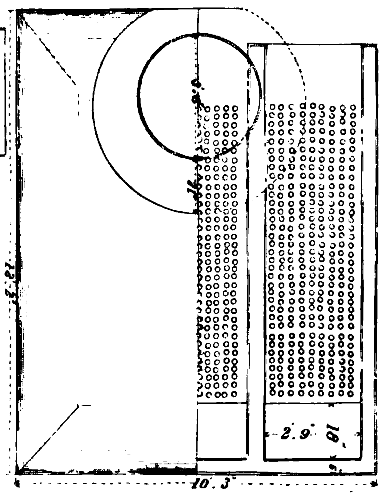

*Enlarged View of the Amory Bridge.*

# *Boiler*

OF THE

**U.S.S. WYANDOTTE.**




*Section on C.D.*

*Top View. Section on A.B.*

| | |
|---|---|
| Total number of tubes, | 933. |
| Length of tubes between plates, | 2 feet 8 inches. |
| External diameter of tubes, | 2 " |
| Length of the space occupied by the tubes of each furnace, | 8 feet 5 " |
| Number of rows of tubes lengthwise the furnaces, | 35. |
| Number of rows of tubes crosswise the furnaces, | 9. |
| Distance between centres of tubes lengthwise the furnaces, | 2·912 " |
| Distance between centres of tubes crosswise the furnaces, | 3½ " |
| Aggregate area of heating surface in the tubes, measured on the external circumference, | 1302·67 square feet. |
| Aggregate area of all water heating surface except in tubes, | 690·68 " |
| Total area of all water heating surface, | 1993·35 " |
| Total area of grate surface, | 58⅝ " |
| Aggregate calorimeter or cross area between tubes for draught, | 10· " |
| Total steam room, | 526· cubic feet. |
| Weight of fresh water in boiler filled to 1 foot above top tube-plate, | 28,097· pounds. |
| Ratio of heating to grate surface, | 37·172 to 1·000. |
| Ratio of grate surface to calorimeter, | 5·362 " |

The furnace door opening was 18 inches wide, semicircular on top, and 16 inches in extreme height. The door was of cast iron fitted very accurately to the shell of the boiler, and was pierced with ten holes of one inch diameter evenly distributed over the surface. Inside the door and closely fitted to it was a *box*-formed lining plate of ⅜-inch thick wrought iron, pierced with eighty-three holes of ⅜-inch diameter evenly distributed over the surface. All the fittings were made as close as practicable, in order to prevent any access of air through the furnace door opening that did not pass through the holes in the lining plate. The height from the bottom of the dead plate at the front of the furnace to the bottom of the ash-pit was 20 inches. Height from bottom of grate-bars at back of furnace to bottom of ash-pit was 15 inches.

The grate-bars were of cast iron and in two lengths. Thickness of bar on top 1 inch; on bottom ½-inch. Depth of grate-bar at ends 2½ inches; at centre 5 inches. Width of air space between bars ⅝-inch. The top of the bar was grooved out its whole length.

Mr. Amory was authorized to construct and apply his invention to the boiler just described, and to superintend the experiments with it, regulating the mode of firing, rapidity of the combustion, &c., in whatever manner he considered best for developing the merits of his invention. He availed himself of this authority to construct a thing (Figs. 1 and 2, Plate 12,) quite different from his invention, both in form and application. Instead of being a combustion chamber it was simply a hollow bridge wall of boiler plate, open at bottom and concave towards the furnace, and instead of being placed behind the regular bridge wall was simply set upon the grate-bars immediately in front of it. The atmosphere had free access to the interior of this hollow bridge wall through the openings between the grate bars. An open wrought iron tube of 1⅝ inch internal diameter was brought from the front of the ash-pit, carried along beneath the grate-bars, and turned up into the hollow bridge wall. It was wholly useless and seems to have been put in simply because a similar tube is mentioned in the Letters Patent.

It has been stated that the front of the hollow bridge was concave towards the furnace; the upper part curved over to a horizontal tangent; the back of the bridge was also concave towards the furnace, but of flatter curvature than the front. At the top of the bridge the back and front plates nearly joined, but

they diverged as they approached the bottom, making the bridge a spherical triangle in cross section with its base resting on the grate-bars. The plates composing the front and back of the bridge were separated at top by a space $\frac{3}{8}$-inch wide and extending across the width of the furnace, making a length of air opening, excluding the space occupied by the five $1\frac{1}{2}$ inch diameter socket bolts connecting the plates, of $25\frac{1}{2}$ inches. The front of the bridge was perforated with fourteen holes, each of $\frac{3}{16}$-inch diameter and equally spaced over the surface. The total air opening from the bridge into the furnace was thus 10 square inches, and the air was discharged nearly horizontally over the solid fuel on the grate-bars. The exterior width of the hollow bridge at bottom was 9 inches, and the space between it and the regular bridge wall was filled with fire-clay. The extreme height of the hollow bridge above the grate was 15 inches, and it was composed of $\frac{5}{16}$-inch thick boiler plate stiffened by cross braces. The hollow bridges as thus applied, of course, reduced the effective length of the grate-bars 9 inches.

From this description it will be perceived that the thing experimented with as Mr. AMORY'S invention was simply a hollow boiler plate bridge wall admitting air to the back of the furnace through a slit at its top $\frac{3}{8}$-inch wide and extending across the furnace, and through fourteen holes of $\frac{3}{16}$-inch diameter each. The concavity of the front must have been unproductive of any effect; in fact, one-half of it was occupied by the solid fuel. The air thus admitted to the furnace had, of course, whatever temperature it could obtain from the plates of the bridge, but that temperature, being wholly at the expense of the fuel, could not be productive of increased economic effect. In substance, then, the Amory bridge, as applied, was simply a means for introducing heated air at the back of the furnace, the heating of the air being effected not by waste heat, but directly by the fuel on the grate-bars and wholly at its expense.

## MANNER OF MAKING THE EXPERIMENTS.

The experiments made with the Amory bridge applied, and without it, hereinafter detailed, were conducted in the following manner, namely:—

The boiler was placed in a temporary shed, and well covered with thick new felt stitched to canvass. To the manhole—which was an ellipse having conjugate and transverse diameters of 15 and 11 inches—was fitted a temporary steam escape-pipe of sheet iron 14 inches in diameter. The manhole was in the front of the boiler near the crown, and the elbow of the escape-pipe was made slightly dropping with a small drip-pipe at its lowest part, in order that the steam condensed in the pipe should not run back into the boiler to be again evaporated. A glass gauge was attached to the boiler to determine the water level. The boiler was perfectly new, double riveted, and had been found tight under a hydrostatic pressure of 35 pounds per square inch. As it was accessible to observation in every part, outside and inside, the slightest leakage could not have escaped detection.

The feed-water was measured in a wooden tank lined with sheet lead; the horizontal dimensions of the tank were 7 feet $11\frac{3}{4}$ inches by 5 feet $5\frac{3}{4}$ inches, and it was found by careful trial that to contain 170 cubic feet it required to be filled to the depth of 3 feet $10\frac{3}{8}$ inches. At this point a mark was made and the tank, each time, was exactly filled to it.

The tank was situated above the top of the boiler to which it was connected by a pipe with a stop-cock. The cock being set to the proper opening, the water was fed to the boiler by gravity, and with a little attention in exactly the quantity evaporated, so that its level in the boiler remained nearly constant. The tank was filled through a hose by gravity from the Navy Yard reservoir and the filling occupied but a few seconds.

Care was taken that all the coal consumed, except that for the last two experiments, should be from the same lot and in the same condition as nearly as could be judged. It was Blackheath anthracite from Pennsylvania, of excellent quality, in small lumps and free from dust. It was not selected but taken from the heap as delivered by the contractor for the regular government supplies. The coal consumed in the last two experiments was first quality Broad Top semi-bituminous, also from Pennsylvania, not selected, but, like the anthracite, taken from the heap as delivered by the contractor.

Every pound of the coal burned was accurately weighed, as was also the refuse in ashes, &c., in a *dry* state. All the weighing was done with the same scales.

Before commencing an experiment the boiler was filled to a given mark on the glass gauge and the water brought to the boiling point with wood, which was then allowed to burn out to the few embers necessary for kindling the coal. The water level on the gauge was now marked, the time noted, and the firing with coal being commenced the experiment was considered as begun. Each experiment was continued exactly seventy-two consecutive hours, and the fires were so regulated towards its close as to be nearly burned out. The moment the time expired the contents of the furnaces were drawn and the unconsumed coal carefully picked out, weighed, and deducted. Care was also taken that at the end of the experiment the water level on the gauge should be exactly the same as at the beginning.

Throughout each experiment the coal was regularly fed to the furnaces, a given quantity being weighed out each hour and burned. Care was also had in keeping the thickness of the bed of coal on the grates constant; and in firing and cleaning fires in the same manner and at equal intervals.

During each experiment a tabular record was kept in which was entered each hour the number of pounds of coal consumed, the exact time at which each tankful of water was emptied, the height of the barometer, and the temperature of the external atmosphere in the shade, of the water in the tank, and of the products of combustion as they emerged from the tubes into the boiler uptake. This latter temperature was obtained as follows, namely:—

A deep copper pot of $3\frac{1}{2}$ inches diameter was filled with oil and suspended in front of the tubes of the middle furnace and as near them as possible without contact. In this pot a thermometer was placed and its indications were hourly recorded as the temperature of the products of combustion leaving the boiler.

At the close of each experiment the furnaces and tubes were swept, and the soot, ashes, &c., collected were added to the quantity of refuse.

The state of the weather was also noted each watch of four hours.

In making the experiment with the Amory bridge recorded in column A of the following Table, all the conditons were regulated by Mr. AMORY'S agent. Five out of the ten one-inch diameter holes in the furnace door were closed, the damper in the chimney was partially closed, and the ash-pit opening was reduced at the front from the height of 20 inches to 6 inches, the closing being made from the bottom up. The aggregate air opening into each furnace through the furnace door was 3·927 square inches; and through the bridge 10 square inches, making a total of 13·927 square inches. The length of the fire grates in use was 5 feet 9 inches, and the area of grate surface in each furnace 15·812 square feet; the proportion of air opening to grate surface would consequently be $\left(\frac{13{\cdot}927}{15{\cdot}812}=\right)$ 0·881 square inch per square foot of grate. The area of ash-pit opening in front was $(33 \times 6 =)$ 198 square inches.

The experiment with the regular arrangement of the furnace made to compare with that with the Amory bridge is recorded in column B of the following Table. In making this experiment care was

taken that it should, in essentials, be strictly comparable. As nearly as possible the same weight of coal was burned per hour, but being consumed on a grate 9 inches longer, its combustion was proportionately slower, being at the rate of 8·163 pounds per square foot of grate per hour instead of 8·957 pounds; but the thickness of the fires was kept the same in both cases. The weight of combustible consumed in equal time, in proportion to the heat absorbing surface of the boiler, was almost exactly the same, being as 27261 to 27378. As in the experiment with the Amory bridge, so in this, the damper in the chimney was kept partially closed, but all of the ten one-inch diameter holes in the furnace doors were carried open. The ash-pit opening in front, too, was likewise reduced from the height of 20 inches to $2\frac{3}{4}$ inches, closing it from the bottom up, as that reduction was found necessary to restrict the combustion to the same weight of coal in equal times as with the Amory bridge. The area of the ash-pit opening was consequently $(33 \times 2\frac{3}{4} =)$ $90\frac{3}{4}$ square inches for each furnace. The aggregate air openings into each furnace through the furnace door was 7·854 square inches, and as the area of the fire grate was 17·875 square feet, the proportion of air opening to grate surface was $\left(\frac{7{\cdot}854}{17{\cdot}875} = \right)$ 0·4394 square inch per square foot of grate. The similar *proportion* with the Amory bridge was a little more than double this; but the *absolute* areas of opening compared as 13·927 for the experiment with the Amory bridge to 7·854 for the experiment with the regular arrangement of the furnace.

The boiler had been constructed without reference to these or any other experiments, and with the proportions, and perforated furnace doors, adopted for its type using anthracite; and it was considered proper to test it as built against the Amory bridge as arranged for best effect by the patentee, burning in each case the same weight of coal per hour, and carrying fires of equal thickness; the rapidity of the combustion and the thickness of the fires being chosen by the patentee's agent.

After the completion of the experiment (A) with the Amory bridge, and of the comparable experiment (B) with the regular arrangement of furnace, it was considered desirable to make another with the regular arrangement of furnace and under as nearly the same conditions as could be commanded as those of the experiment recorded in column B of the following Table, with the single exception of having all the holes in the furnace doors closed, so that no air could be admitted to the furnaces except through the grate bars. The ash-pit opening in front was reduced from 20 inches to 2 inches, instead of $2\frac{3}{4}$ inches, the chimney damper, as before, being partly closed. The thickness of the fires was 5 inches as before, and the weight of combustible consumed in equal times compared as 26934 to 27378. The object of this experiment was, of course, to determine the economic value of the admission of air to the furnaces over the solid fuel under the two modes of experiments A and B, compared with its exclusion. The data and results of this experiment will be found in column C of the following Table.

Advantage was taken of the convenient opportunity afforded by the apparatus for the foregoing experiments, to make the four others recorded in columns D, E, F, and G.

The experiments in D and E were made with the anthracite used in the previous ones. The object was to determine the effect upon the economic evaporation of employing the maximum combustion possible with the fires $8\frac{1}{2}$ inches thick, and the holes in the furnace doors first open and then closed. In column D will be found the results with the holes in the furnace doors open; and in column E the results with these holes closed. In these two experiments, which are strictly comparable, the entire ash-pit opening of 20 inches high was used with the chimney damper wide open.

The experiments recorded in columns F and G were made with the Broad Top semi-bituminous coal to

**TABLE**—CONTAINING THE DATA AN]S. STEAMER "WYANDOTTE," HAVING ITS TUBES OVER THE FUNGEMENT OF FURNACE ADOPTED IN THE U. S. NAVAL SERVICALSO, TO DETERMINE THE RELATIVE ECONOMIC EVAPORATION ALS FROM PENNSYLVANIA), UNDER DIFFERENT CONDITLES IN THE FURNACE DOORS.

| NUMBER OF LINE. | | | BROAD TOP SEMI-BITUMINOUS COAL. | |
|---|---|---|---|---|
| | | | REGULAR ARRANGEMENT OF FURNACE. | |
| | | | MAXIMUM COMBUSTION. | |
| | | s. | Holes open in furnace doors. F | Holes closed in furnace doors. G |
| 1 | | Date of commencing the experiment, | 12·30 P. M. Dec. 26. | 11·35 A. M. Jan. 2. |
| 2 | | Date of ending the experiment, | 12·30 P. M. Dec. 29. | 11·35 A. M. Jan. 5. |
| 3 | TOTAL QUANTITIES. | Duration of the experiment in consecutive hours, | 72· | 72· |
| 4 | | Number of cubic feet of water evaporated, | 6,316·72 | 6,436·93 |
| 5 | | Number of pounds of water evaporated, | 394,062·26 | 401,548·63 |
| 6 | | Number of pounds of coal consumed, | 44,880· | 45,120· |
| 7 | | Number of pounds of refuse from the coal, in ashes, | 5,211· | 3,866· |
| 8 | | Number of pounds of combustible consumed, | 39,669· | 41,254· |
| 9 | | Per centum of the coal in refuse, | 11·61 | 8·57 |
| 10 | MEAN QUANTITIES. | Temperature in degrees Fahr. of the external atmos shade, | 24·7 | 20·1 |
| 11 | | Temperature in degrees Fahr. of the water in the tai | 42·4 | 40·0 |
| 12 | | Temperature in degrees Fahr. of the products of con tering the chimney, | 243·4 | 285·0 |
| 13 | | Barometer, | 30·3 | 30·1 |
| 14 | | Thickness of the bed of coal upon the grates, in inch | 8·5 | 8·5 |
| 15 | | Pounds of coal consumed per hour per square foot o face, | 11·624 | 11·686 |
| 16 | | Pounds of combustible consumed per hour per square surface, | 10·274 | 10·685 |
| 17 | EVAPORATION. | Total number of pounds of water evaporated from ter 100° Fahr., | 415,113·024 | 423,895·166 |
| 18 | | Total number of pounds of water evaporated from ter 212° Fahr., | 463,229·618 | 473,034·801 |
| 19 | | Pounds of water evaporated from temperature of 10 one pound of coal, | 9·249 | 9·395 |
| 20 | | Pounds of water evaporated from temperature of 21 one pound of coal, | 10,321 | 10·484 |
| 21 | | Pounds of water evaporated from temperature of 10 one pound of combustible, | 10,465 | 10·275 |
| 22 | | Pounds of water evaporated from temperature of 21 one pound of combustible, | 11·677 | 11·466 |
| 23 | | State of the weather, udy ne; ths lear | Moderate breezes and clear sky. | Moderate breezes and clear sky. |

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determine its economic evaporation under the two conditions of having the holes in the furnace doors first open and then closed. With the exception of the difference in the kind of coal, all the circumstances were like those of the experiments in columns D and G. The fires were carried 8½ inches thick, and the whole ash-pit opening was employed together with a wide damper in order to give the maximum combustion. In the experiment recorded in column F the holes in the furnace door were open; in that recorded in column G these holes were closed.

## EXPLANATION OF THE TABLE CONTAINING THE DATA AND RESULTS OF THE EXPERIMENTS.

In the following table will be found arranged in columns all the observed data and the calculated results of the experiments. For facility of reference the columns are designated by letters, and the lines containing the quantities by numbers. For the same purpose the quantities have been arranged in groups.

The experiments recorded in columns A and B were designed to be strictly comparable for showing the relative economic evaporation by the regular arrangement of furnace with perforated door adopted in the U. S. Naval Service for vertical water tube boilers with the tubes over the furnaces burning anthracite (column B), and by the same with the Amory bridge applied (column A).

The experiment in column C is strictly comparable with that in column B, and shows the effect upon the economic evaporation, *ceteris paribus*, of the anthracite by closing the holes in the furnace doors with thin fires (5 inches thick) and a slow combustion (about 8 pounds of coal per square foot of grate per hour).

The experiment in column D was made to show the effect upon the economic evaporation of the anthracite by increasing the thickness of the fires from 5 inches to 8½ inches, with the holes in the furnace doors open and the rate of combustion the maximum attainable by the boiler (about 11½ pounds of coal per square foot of grate per hour). This experiment, as regards the effect of increased thickness of fire and rate of combustion, is strictly comparable with that in column B.

The experiment in column E was made under precisely the same circumstances as that in column D with the exception of having the holes in the furnace doors closed. It was designed to show the effect upon the economic evaporation of the anthracite of the suppression of air through the holes of the furnace doors under the conditions of carrying the fires at the increased thickness of 8½ inches with a maximum combustion. This experiment is strictly comparable with that in column D.

The experiments recorded in columns F and G were made with Broad Top semi-bituminous coal, but otherwise they were, respectively, precisely like those in columns D and E, the holes in the furnace doors being open in column F and closed in column G.

When experiments F and G are compared together, they show the effect upon the economic evaporation of Broad Top semi-bituminous coal, with fires 8½ inches thick and the combustion at the maximum, of suppressing the admission of air through the holes in the furnace door. When they are compared, respectively, with experiments D and E, they show the relative economic evaporative efficiency of the Blackheath anthracite and the Broad Top semi-bituminous coal under the two conditions of the admission and suppression of air through the holes in the furnace doors.

Lines 1 and 2 show the day and hour of commencing and ending the experiments.

TOTAL QUANTITIES. Line 3 shows the number of hours each experiment was continued without intermission and during which its conduct was maintained as uniform as possible.

Line 4 contains the number of cubic feet of water evaporated from the temperature on line 11, ascertained by previous measurement in the tank.

Line 5 contains the number of pounds of water evaporated from the same temperature. This quantity is calculated from that on line 4, taking the weight of a cubic foot of water at 62° Fahr. to be 62·321 pounds, and modifying it for the temperature on line 11, according to KOPP'S table.

Line 6 contains the number of pounds of coal consumed. Every pound was carefully weighed. The anthracite was from the "Middle Anthracite Field" of Pennsylvania. It was all in clean lumps of medium size, and was very carefully fired. The Broad Top semi-bituminous coal was from the southern part of Pennsylvania, between the Susquehanna river and the foot of the Allegheny mountains. It, too, was in excellent condition and mostly in clean lumps.

Line 7 contains the number of pounds of refuse from the coal in ashes, clinker, and dust, inclusive of the sweepings at the end of each experiment. It was weighed on the same scales as the coal, and in a dry state.

Line 8 contains the number of pounds of combustible consumed. It is the quantity on line 6 less that on line 7.

Line 9 shows the per centum which the quantity on line 7 is of the quantity on line 6.

MEAN QUANTITIES. The quantities on lines 10, 11, and 12 show, respectively, the temperature in degrees Fahr. of the atmosphere in the shade, of the water in the tank, and of the products of combustion as they emerge from the tubes into the boiler uptake. They are each the mean of seventy-two observations at one hour apart.

Line 13 shows the mean height of the barometer. It was almost exactly the same in all the experiments, and indicated a pressure of 14·75 pounds per square inch. It, also, is the mean of seventy-two hourly observations.

Line 14 shows the thickness of the bed of anthracite upon the grates. It was very accurately gauged by rivet marks on the sides of the furnaces, and was kept as uniform as possible.

The quantity on line 15 is obtained by dividing the quantity on line 6 by 72, and, for column A, by 47·437, the number of square feet of grate surface in operation during that experiment; and for the other columns by 53·625, the number of square feet of grate surface in operation during the experiments recorded in them.

The quantity on line 16 was obtained by treating the quantity on line 8 in the same manner.

EVAPORATION. Lines 17 and 18 contain, respectively, the number of pounds of water that would have been evaporated had its temperature in the tank been 100° and 212° Fahr., instead of the temperature on line 11. In calculating the quantities on these lines the total heat of the steam above zero of Fahr. has been taken at 1178·15°.

The quantities on lines 19 and 21 are obtained by dividing the quantity on line 17 by the quantities on lines 6 and 8.

The quantities on lines 20 and 22 are obtained by dividing the quantity on line 18 by the quantities on lines 6 and 8.

Line 23 shows the state of the weather during each experiment.

## DISCUSSION OF THE RESULTS.

In comparing the results of the experiments, the evaporative efficiency of the coal will be taken to be correctly measured by the number of pounds of water evaporated per pound of combustible (i. e. the remainder of the coal after deducting the refuse) from the temperature of 100° Fahr., (line 21). The pound of *combustible* is preferred to the pound of *coal* for this purpose, as it is obvious the slight difference in the per centum of refuse was a purely accidental matter which might easily have been reversed on a repetition of the experiments.

And, first, as regards the effect, *ceteris paribus*, produced by the AMORY BRIDGE. Comparing in columns A and B, the economic evaporation, we find it to have been with the Amory Bridge 11·529, and with the regular furnace 12·425 pounds of water per pound of combustible of the Blackheath anthracite. Assuming the evaporation with the regular furnace as unity, it appears *the application of the Amory Bridge caused a loss of* $\left(\frac{12{\cdot}425 - 11{\cdot}529 \times 100}{12{\cdot}425} = \right)$ 7·21 *per centum.*

Second, as regards the effect, *ceteris paribus*, produced on the economic evaporation of the anthracite by the suppression of air through the holes in the furnace doors, when the fires are carried thin, (5 inches thick) and with a combustion (about 8 pounds of coal per square foot of grate per hour) equal to about three-fourths of the maximum. It appears, comparing the results in columns B and C, and assuming the evaporation with the holes open for unity, that *the suppression of air entering through them caused a loss of* $\left(\frac{12{\cdot}425 - 11{\cdot}744 \times 100}{12{\cdot}425} = \right)$ 5·48 *per centum.*

Again, when the anthracite was burned at the maximum rate (columns C and D, about 11 pounds per square foot of grates per hour) with fires increased to 8½ inches thickness, it appears assuming for unity the evaporation with the holes in the furnace doors open, (column C) that *the suppression of the air admission through them caused a loss of* $\left(\frac{11{\cdot}123 - 10{\cdot}739 \times 100}{11{\cdot}123} = \right)$ 3·45 *per centum.*

It will be here observed that although the *per centum* of gain by the admission of air through the holes in the furnace door was *less* with the thicker fires and increased rate of combustion, yet *the effect of the admission of the same quantity of air in equal time through these holes was sensibly the same in proportion to weight of coal consumed in equal time;* for example: the gain with the thin fires and slow combustion we have seen to have been 5·48 per centum, while with the thicker fires and maximum combustion it was 3·45 per centum. Now the rates of combustion in the two cases were nearly as 8 and 11; hence 11 : 5·48 :: 8 : 3·98.

As regards the effect produced upon the economic evaporation of the Broad Top semi-bituminous coal (columns F and G) by the suppression of the air admission through the holes in the furnace doors, with the fires carried 8½ inches thick and the combustion at the maximum, (about 11⅝ pounds of coal per square foot of grates per hour) we find, assuming the evaporation with the holes open for unity (column F) *that closing them caused a loss of* $\left(\frac{10{\cdot}465 - 10{\cdot}275 \times 100}{10{\cdot}465} = \right)$ 1·81 *per centum.*

Under the same conditions we have seen the loss with the Blackheath anthracite to have been 3·45 per centum, whence it appears that *the admission of air through holes in the furnace doors was more beneficial*

*with the anthracite than with the semi-bituminous coal;* a result contrary to the popular belief, but which has in several cases been verified by the writer.

It appears probable from these experiments that a larger admission of air through holes in the furnace doors would have been beneficial; and they show, also, the greater economic effect to be obtained from anthracite by burning it in thin fires.

The temperature of the products of combustion entering the chimney (line 12) appears to have been very nearly the same whichever the coal or whatever the quantity burned per hour, showing the area of the heat absorbing surface to have been sufficient for maximum economy for at least the maximum rate of combustion. This temperature, for a mean in the different experiments, was only 39° Fahr. above that of the steam within, showing the tube surface to have possessed a very high efficiency of heat absorbing power.

The fact, however, must not be overlooked that when the evaporation is effected under the working pressure of the boiler—say 20 pounds per square inch above the atmosphere—instead of under the atmospheric pressure, the products of combustion will emerge at at least as much higher temperature as the temperature of the steam of 20 pounds pressure exceeds that of the steam of the atmospheric pressure, (about 47° Fahr.) which in conjunction with the greater amount of heat required to evaporate a given weight of steam under the increased pressure, would slightly reduce the economic evaporation by the fuel.

The relative economic evaporative efficiency of the anthracite and of the semi-bituminous coal will be determined by comparing the results of the experiments recorded in columns D and F, whence, assuming unity for the evaporation by the anthracite per pound of combustible, it appears that the semi-bituminous coal was inferior by $\left(\frac{11{\cdot}123 - 10{\cdot}465 \times 100}{11{\cdot}123} = \right)$ 5·92 per centum of the anthracite.

It must not be forgotten that the results of these experiments, like those of all others of the kind, are strictly true only for the proportions and type of boiler employed, for the species of coal used, and for the conditions described. Variations in any of these might vary the results, both relative and absolute.

The composition of the Blackheath anthracite and its mode of action in the furnace are so well known as to need no particular description here.

The Broad Top semi-bituminous coal is very similar to the well known Cumberland, differing only in being slightly less bituminous. It can be substituted in smiths work for the Cumberland, but is not quite equal to it for that purpose. In the furnace it burns freely and forms into cakes of weak coherence; its agglutination, however, is very slight, and it does not intumesce or give out tarry matter. Upon making a fresh fire, and when breaking up the cakes, smoke is emitted in large quantities; at other times the amount of smoke is inconsiderable.

# EXPERIMENTS

MADE WITH

# THE BOILER OF THE U. S. STEAMER

# "UNDERWRITER."

# EXPERIMENTS

MADE WITH

# THE BOILER OF THE U. S. STEAMER "UNDERWRITER,"

TO DETERMINE ITS EVAPORATIVE EFFICENCY WITH ANTHRACITE, AND WITH CUMBERLAND SEMI-BITUMINOUS COAL.

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ADVANTAGE was taken of the construction of a new boiler for the U. S. Steamer "UNDERWRITER," to determine its evaporative efficiency with the Pennsylvania anthracite from Harvey's Mine, brought down the Susquehanna river, and the Cumberland semi-bituminous coal from Georges' Creek, Maryland.

The experiments were made at Baltimore on the boiler in the vessel at the dock. It was a perfectly new boiler, and had been proven water-tight under a hydrostatic pressure of 60 pounds per square inch. It was thoroughly covered with new thick felt stitched to canvass, over which was a coating of sheet lead for the steam chimney and the top of the shell to the vertical of the sides.

The evaporation was performed under the atmospheric pressure; the steam escaping through a hole of 12 inches diameter placed 6 inches below the top of the steam chimney (the hole for the engine steam-pipe) to which a short pipe of the same diameter had been fitted. This pipe turned down, so that none of the steam condensed in it could return to the boiler for re-evaporation.

The number of cubic feet of water evaporated was ascertained by measurement in an iron tank 69½ by 82½ inches, which was filled each time to exactly 58½ cubic feet. The tank was placed upon the wharf above the level of the boiler. It was filled by gravity from the city hydrant through a hose and stop-cock, and delivered in the same manner into the boiler through a hose provided with a stop-cock, and discharging into the safety-valve chamber. There was required just five minutes to fill the tank. The whole apparatus was in open view, and every precaution was taken to insure against the possibility of the least loss by leakage. A thermometer suspended within the tank near its bottom gave the temperature of the feed-water.

A glass water gauge attached to the boiler showed the precise water level within, which was carried constantly at the height of 8 inches above the top flues during the experiments.

The temperature of the products of combustion in the uptake was ascertained from a mercurial thermometer immersed in fine sand contained in a copper pot.

In commencing an experiment, the water was brought to the boiling point by a fire of pine wood, which was then allowed to burn down to the few embers required for the ignition of the coal; the height of the water being regulated to 8 inches above the top of the flues, the coal was now fired and the experiment held to commence. During its continuance every pound of coal fed to the furnaces was carefully weighed, and it was fired very uniformly, a given weight being fed in during each hour. The fires were carried 7 inches thick, and kept uniformly spread; they were not forced, but neither were they at all retarded by the use of any damper, and the rate of combustion is a fair average of what the boiler can maintain. The air-holes in the furnace doors were kept open during the whole time. The cleaning of the fires was performed at regular intervals, and the whole of the refuse in ashes, clinker, &c., was carefully weighed in the *dry* state every four hours. At the close of the experiment, the fires were burned out as nearly as possible, the furnaces drawn at the moment the duration of the experiment expired, and the unconsumed coal picked out, weighed, and deducted. At the end of the experiment the water in the glass gauge was left precisely at the same point as at the commencement.

The experiment with the anthracite continued uninterruptedly 48 hours, that with the semi-bituminous coal 24 hours. They were limited by the urgent necessity for the use of the vessel, and were made by her Engineer Department. The Engineers stood regular watches of four hours and attended personally to the weighing of the coal and ashes, to the filling and emptying of the tank, and to recording the observations. They kept a tabular record in which was noted hourly the weight of coal supplied to the furnaces during that hour; the height of the barometer; the temperatures of the external atmosphere in the shade, of the fire room, of the water in the tank, and of the products of combustion in the uptake. There were also noted the exact minute when each tankful of water was emptied; and the state of the weather; the weight of ashes was recorded every four hours. The totals and means of these quantities will be found in the table hereinafter given of the results of the experiments.

The coals were not picked, but were taken in the merchantable condition as furnished by the contractors.

In the following will be found a description, and the dimensions of the boiler.

## BOILER.

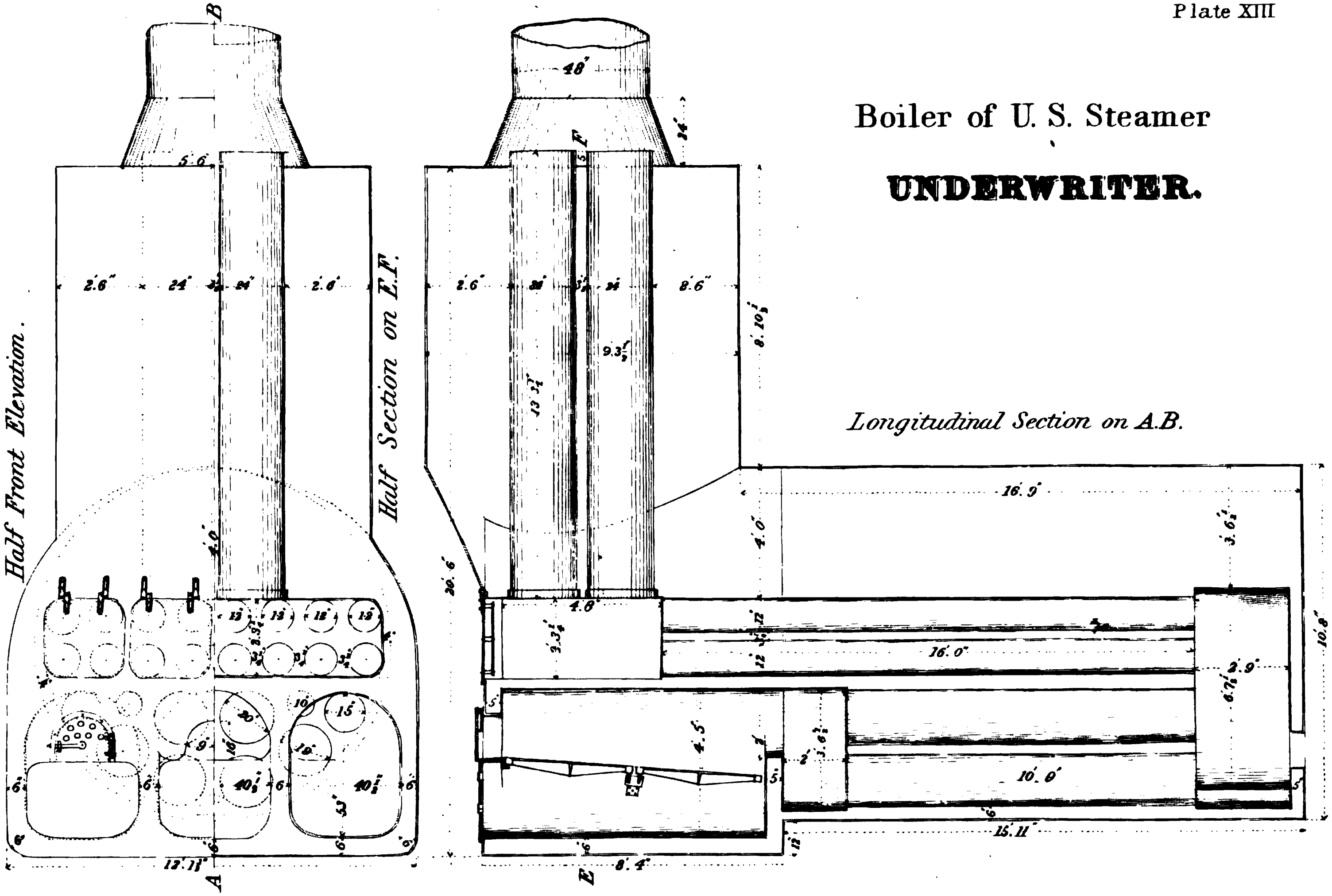
The boiler is of the kind known as the single return ascending flue. The products of combustion, after passing from the furnaces over the bridge walls, enter the first smoke-connexion, which is in common for all the furnaces; from this connexion these products pass through horizontal flues, immediately behind the furnaces, to the second smoke connexion, which is at the back of the boiler, and, like the first, is also in common for all the furnaces. From the upper part of the second smoke connexion the products of combustion return through horizontal flues above the level of the furnaces to the uptake, which is in common at the front end of the boiler, and from which there rise vertically superheating flues that debouch into the chimney.

The shell of the boiler consists of a front, rectangular in plan, vertical on the sides and semicircular on the top, containing the furnaces. Width of front 12 feet 1½ inch; extreme height of front 11 feet 8 inches; extreme length of front 8 feet 4 inches. Behind this front there is a cylindrical part of 10 feet 8 inches

Plate XIII

Boiler of U. S. Steamer

**UNDERWRITER.**



diameter, and 15 feet 11 inches length; making the total length of the shell of the boiler 24 feet 3 inches. The top of the front and the top of the cylindrical part are on the same level. The superheating flues, which form the lower part of the chimney and are at the front end of the boiler immediately over the furnaces, are surrounded by a cylindrical steam chimney or drum of 9 feet 3½ inches diameter and 8 feet 10½ inches height above the shell of the boiler. From the top of this drum the steam is carried to the engine through a pipe of 12 inches diameter.

The furnaces are three in number, 40½ inches wide and 7 feet 6 inches long; they have semicircular crowns; the height from bottom of ash-pit to crown of furnace is 53 inches; the corners of the ash-pits are rounded on a radius of 12 inches; the width of water bottom and water legs is 6 inches. Each furnace has a door with an opening of 18 inches in width by 16 inches in height, semicircular on top. Each door is perforated with eleven holes of 1¼ inch diameter; and its lining is perforated with three hundred and fifty-one holes of ¼-inch diameter. The water spaces at the front and back of furnaces are 5 inches wide.

The first smoke connexion is flat on top, and on the sides and bottom it is concentric with the circular shell of the boiler; the width of the water space between it and the shell is 5 inches. The extreme height of the connexion is 42½ inches, and its extreme breadth, lengthways the boiler, is 24 inches.

The lower flues are ten in number, and 10 feet 9 inches in length; four of them are 19 inches in inner diameter, two are 20 inches in diameter, two are 15 inches in diameter, and two are 10 inches in diameter.

The second smoke connexion is flat on top, and on the sides and bottom it is concentric with the circular shell of the boiler; the width of the water space between it and the shell is 5 inches. The extreme height of the connexion is 73½ inches, and its extreme breadth lengthways the boiler, is 33 inches.

The upper flues are in two tiers of eight flues each, one immediately over the other; they are 12 inches in inner diameter, and 16 feet in length. Water spaces between the flues, including thicknesses of metal 3¼ inches.

The uptake is flat on top, bottom and sides. Its extreme height is 27¼ inches, and its extreme breadth lengthways the vessel, is 56 inches. Water space between bottom of uptake and crown of furnace 5 inches including thicknesses of metal. The front of the uptake is fitted with four doors for sweeping the flues; the door openings are separated by water legs, 3¼ inches wide by 5 inches deep.

The superheating flues are four in number; they are 2 feet in inner diameter, and 12 feet 10¾ inches in height. The least steam space between them is 3½ inches.

The water bottom of the boiler, and the bottom of the ash-pits are composed of $\frac{7}{16}$-inch thick iron plate. The 10 and 12 inches diameter horizontal flues are of ¼-inch thick plate, the remaining horizontal flues are of $\frac{5}{16}$-inch thick plate. All the other parts of the boiler are of ⅜-inch thick plate. The flat spaces are braced every 10 inches. All the seams not in contact with the fire are double riveted.

The following are the principal dimensions and proportions of the boiler, namely:—

| | |
|---|---|
| Extreme height, exclusive of steam chimney, . . . | 11 feet 8 inches. |
| Extreme breadth, . . . . . . | 12 " 1½ " |
| Extreme length, exclusive of steam chimney, . . . | 24 " 3 " |
| Number of furnaces, . . . . . . | 3. |
| Breadth of furnaces, . . . . . . | 3 feet 4½ inches. |
| Length of fire grates, . . . . . . | 7 " 6 " |

| | |
|---|---|
| Aggregate area of grate surface, | 75·9375 square feet. |
| Aggregate area of heating surface in the three furnaces, | 186·00 " |
| Aggregate area of heating surface in the first smoke connexion, | 79·23 " |
| Aggregate area of heating surface in the ten lower flues, | 466·82 " |
| Aggregate area of heating surface in the second smoke connexion, | 169·67 " |
| Aggregate area of heating surface in the sixteen upper flues, | 804·48 " |
| Aggregate area of heating surface in the uptake, | 111·50 " |
| Total water heating surface, | 1,817·70 " |
| Aggregate area of steam superheating surface in the four superheating flues, | 326·56 " |
| Weight of the boiler, exclusive of grate-bars and smoke-pipe, but inclusive of doors and plates, | 82,322· pounds. |
| Weight of water in the boiler to 12 inches above flues, | 51,500· " |
| Diameter of the chimney, | 4 feet. |
| Height of the chimney above the level of the grates, | 52 " |
| Aggregate area of the spaces above bridge walls for draught, | 16·93 square feet. |
| Aggregate area of the lower flues, for draught, | 15·78 " |
| Aggregate area of the upper flues, for draught, | 12·50 " |
| Aggregate area of the superheating flues, for draught, and of the chimney, | 12·56 " |
| Capacity of steam room in shell of boiler, | 570· cubic feet. |
| Capacity of steam room in steam chimney or drum, | 508· " |
| Capacity of steam room in shell and steam drum, | 1,078· " |
| Proportion of heating to grate surface, | 23·937 to 1·000. |
| Proportion of grate surface to area of spaces above bridge walls, | 4·487 " |
| Proportion of grate surface to area of lower flues, | 4·812 " |
| Proportion of grate surface to area of upper flues, | 9·492 " |
| Proportion of grate surface to area of superheating flues, | 6·046 " |

With the boiler described and the experiments conducted in the manner related, the following results were obtained, namely:—

## DATA AND RESULTS OF THE EXPERIMENT MADE WITH THE BOILER OF THE U. S. STEAMER "UNDERWRITER," TO DETERMINE ITS EVAPORATIVE EFFICIENCY WITH SUSQUEHANNA ANTHRACITE FROM HARVEY'S MINE.

| | | |
|---|---|---|
| | Date of commencing the experiment, | Noon, July 20, 1862. |
| | Date of ending the experiment, | Noon, July 22, 1862. |
| | State of the weather, | Light breezes and cloudy. |
| TOTAL QUANTITIES. | Duration of the experiment in consecutive hours, | 48 |
| | Cubic feet of water evaporated, as measured in the tank, | 5,592· |
| | Pounds of water evaporated from the temperature of the water in the tank, | 348,138·711 |
| | Pounds of anthracite consumed, | 39,950· |
| | Pounds of refuse from the anthracite in ashes, clinker, &c., | 4,542· |
| | Pounds of combustible consumed, | 35,408· |
| | Per centum of the anthracite in ashes, clinker, and dust, | 11·37 |
| MEAN QUANTITIES. | Temperature in degrees Fahr. of the external atmosphere in the shade, | 76·9 |
| | Temperature in degrees Fahr. of the fire room, | 91·5 |
| | Temperature in degrees Fahr. of the feed-water in the tank, | 72·0 |
| | Temperature in degrees Fahr. of the products of combustion in the boiler uptake, | 370·0 |
| | Barometer, | 29·89 |
| | Pounds of coal consumed per hour per square foot of grate surface, | 11·169 |
| | Pounds of combustible consumed per hour per square foot of grate surface, | 9·922 |
| EVAPORATION. | Pounds of water evaporated from the temperature of 100° Fahr., | 357,180·939 |
| | Pounds of water evaporated from the temperature of 212° Fahr., | 398,591·507 |
| | Pounds of water evaporated by one pound of anthracite from the temperature of 100° Fahr., | 8 940 |
| | Pounds of water evaporated by one pound of anthracite from the temperature of 212° Fahr., | 9·977 |
| | Pounds of water evaporated by one pound of combustible from the temperature of 100° Fahr., | 10·087 |
| | Pounds of water evaporated by one pound of combustible from the temperature of 212° Fahr., | 11·257 |

DATA AND RESULTS OF THE EXPERIMENT MADE WITH THE BOILER OF THE U. S. STEAMER "UNDERWRITER," TO DETERMINE ITS EVAPORATIVE EFFICIENCY WITH CUMBERLAND SEMI-BITUMINOUS COAL FROM THE BARTON MINES OF GEORGES' CREEK, MD.

| | | |
|---|---|---|
| | Date of commencing the experiment, | 1 P. M. July 22, 1862. |
| | Date of ending the experiment, | 1 P. M. July 23, 1862. |
| | State of the weather, | Light breezes and cloudy. |
| TOTAL QUANTITIES. | Duration of the experiment in consecutive hours, | 24 |
| | Cubic feet of water evaporated, as measured in the tank, | 3,029· |
| | Pounds of water evaporated from the temperature of the water in the tank, | 188,575·135 |
| | Pounds of semi-bituminous coal consumed, | 20,105· |
| | Pounds of refuse from the coal in ashes, clinker, &c., | 2,652· |
| | Pounds of combustible consumed, | 17,453· |
| | Per centum of the coal in ashes, clinker, and dust, | 13·19 |
| MEAN QUANTITIES. | Temperature in degrees Fahr. of the external atmosphere in the shade, | 68·3 |
| | Temperature in degrees Fahr. of the fire room, | 88·9 |
| | Temperature in degrees Fahr. of the feed-water in the tank, | 72·0 |
| | Temperature in degrees Fahr. of the products of combustion in the boiler uptake, | 412·7 |
| | Barometer, | 30·05 |
| | Pounds of coal consumed per hour per square foot of grate surface, | 11·031 |
| | Pounds of combustible consumed per hour per square foot of grate surface, | 9·576 |
| EVAPORATION. | Pounds of water evaporated from the temperature of 100° Fahr., | 193,472·599 |
| | Pounds of water evaporated from the temperature of 212° Fahr., | 215,901·187 |
| | Pounds of water evaporated by one pound of semi-bituminous coal from the temperature of 100° Fahr., | 9·623 |
| | Pounds of water evaporated by one pound of semi-bituminous coal from the temperature of 212° Fahr., | 10·738 |
| | Pounds of water evaporated by one pound of combustible from the temperature of 100° Fahr., | 11·085 |
| | Pounds of water evaporated by one pound of combustible from the temperature of 212° Fahr., | 12·370 |

From the preceding tables it will be seen that, assuming the evaporation from temperature of 100° Fahr., by the pound of anthracite for unity, the economic result from the semi-bituminous coal was $\left(\frac{9·623-8·940\times 100}{9·623}=\right)$ 7·10 per centum greater.

If, however, we compare the evaporation produced respectively by the pound of combustible of the anthracite, and the pound of combustible of the semi-bituminous coal, from the temperature of 100° Fahr., assuming the former for unity, we shall find the economic result from the latter to be

$$\left(\frac{11·085-10·087\times 100}{11·085}=\right) \text{ 9·00 per centum greater.}$$

# EXPERIMENT

MADE WITH

# THE BOILER OF THE UNITED STATES STEAMER

# "YOUNG AMERICA."

# EXPERIMENT

MADE WITH

# THE BOILER OF THE U. S. STEAMER "YOUNG AMERICA,"

TO DETERMINE ITS EVAPORATIVE EFFICIENCY WITH ANTHRACITE.

---

THE boiler of the U. S. Steamer "YOUNG AMERICA," on which the following experiment was made, was quite new, having never before been used. Previously to being placed in the vessel it had been tested and made perfectly tight under a hydrostatic pressure of 50 pounds per square inch above the atmosphere. After being placed in the vessel it was covered with thick felt stitched upon canvass, in addition to which the steam drum and the top of the boiler to its vertical sides were covered with sheet lead.

The experiment was made at the city of Baltimore on the boiler in the vessel at the wharf. The evaporation was effected under the atmospheric pressure, the steam escaping through a temporary pipe bolted to the opening in the steam drum for the main steam-pipe. This opening was 6 inches in diameter, and in the side of the drum near its top. The escape pipe was of the same diameter; it was made to droop on leaving the drum, and had a hole at its lowest depression for the escape of the water resulting from the condensation of steam in the pipe, none of which re-entered the boiler.

The water evaporated was first measured in an iron tank placed on the wharf. This tank was each time filled to precisely 128·154 cubic feet from the city water works. The water entered, by gravity, through a hose with a stop-cock, and was delivered into the boiler at the safety-valve opening, by gravity, through another hose and stop-cock.

The coal consumed was fair merchantable anthracite from Schuylkill County, Pennsylvania, in lumps, free from dust and dirt.

## MANNER OF MAKING THE EXPERIMENT.

The experiment was conducted by the engineer department of the vessel, and with every precaution to ensure perfect accuracy in the results. It continued seventy-two consecutive hours, during which every pound of anthracite fed to the furnaces, and every pound of refuse withdrawn from them, were carefully weighed. The water in the boiler was carried uniformly at 9 inches above the top of the tubes; and the bed of anthracite on the grates was maintained at 8 inches thickness, kept fairly leveled, and free from holes. The combustion was neither forced nor retarded, but all the anthracite that could be consumed under the conditions was burnt. The fires were cleaned at regular intervals. The boiler was fitted with a glass gauge, and with a mercurial thermometer for denoting the temperature of the gases in the uptake. This thermometer was immersed in fine sand placed in a copper pot suspended in the uptake opposite the centre tube above the middle furnace, and as close to the mouth of the tube as it could be placed without touching. It could be drawn to a small aperture in the uptake door and read without withdrawing it from the boiler; the aperture was closed by a small door.

Steam was first raised by wood, which was then allowed to burn down to the few coals necessary to kindle the anthracite, when, the water level having been adjusted to the proper height, the time was noted, the anthracite fired, and the experiment held to commence. At the close of the experiment the fires were allowed to burn nearly out; and, at the expiration of the seventy-two hours, the water level in the boiler having been adjusted to precisely the same height as at the commencement, they were drawn, and the unburnt coal was carefully separated from the refuse, weighed, and deducted from the total amount of coal expended. All the refuse was weighed in the *dry* state.

During the experiment a tabular record was kept, in which was noted hourly the temperature of the external atmosphere on deck, of the fire room, of the feed-water in the tank, and of the products of combustion in the uptake. Also, the height of the barometer, and the number of pounds of coal fed to the furnaces. In other columns were entered the exact time at which each tankful of water was emptied, the number of pounds of refuse withdrawn from the furnaces and ash-pits, and the state of the weather.

The engineers and firemen of the vessel stood regular watches of four hours, and the former personally attended to the weighing of the coal and refuse, and to the filling and emptying of the tank. They kept the record, and saw that the firing and cleaning was performed with scrupulous regularity. The holes in the furnace doors were kept open during the entire experiment.

The following is a description of the boiler with which the experiment was made, namely:—

## BOILER.

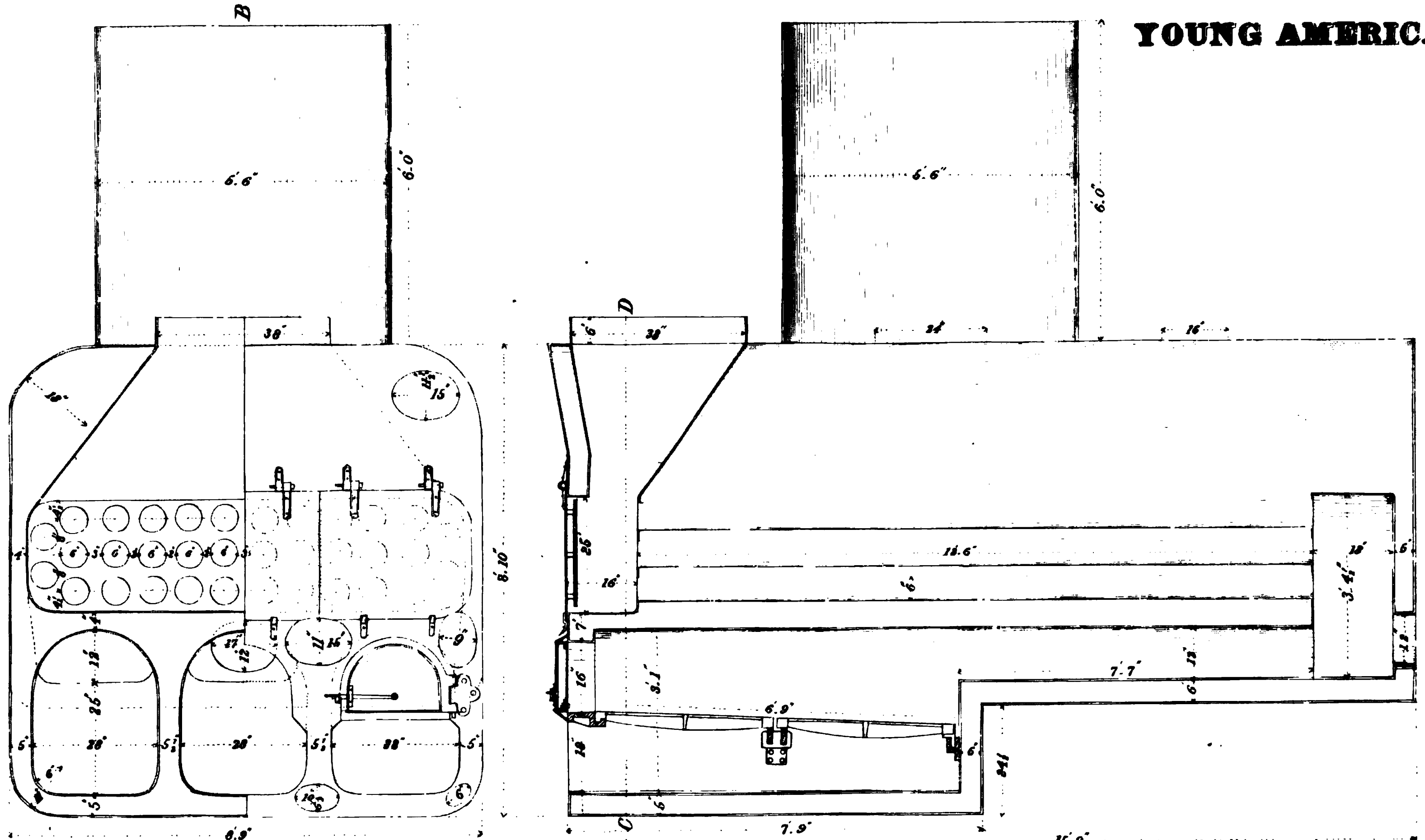
The boiler is of the type with a combustion chamber extended horizontally from each furnace to the after smoke connexion, and horizontal fire tubes returned above the combustion chambers and furnaces to the uptake at the front.

The shell is rectangular, with the corners at the sides rounded. From the back of the furnace portion length for 8 feet to the back of the boiler, the shell is 24½ inches less deep than the furnace portion length of 7 feet 9 inches. The top of the shell for its entire length is in the same horizontal plane.

Plate XIV.

Boiler for U. S. Steamer

**YOUNG AMERICA.**



Half Section on C.D. Half Elevation.

Section on A.B.

The furnaces are three in number, 28 inches wide in the clear, and 6 feet 9 inches in length; they are semicircular on top. Depth from crown of furnace to bottom of ash-pit 37 inches in the clear. The angles of the ash-pits are rounded on a radius of 6 inches. The water legs between the furnaces are 5½ inches wide, including thicknesses of metal; those between the furnaces and the shell, and the water bottom beneath the ash-pits, are 5 inches wide, including thicknesses of metal. The water spaces at the front and back of furnaces are 6 inches wide including thicknesses of metal.

The opening for the furnace door is semicircular on top, 16 inches high and 20 inches wide in the clear. The door is perforated with twelve holes of 1 inch diameter, having the aggregete area of 9·4248 square inches for the admission of air to the furnace above the fuel; it has a box lining plate perforated with thirty holes of ¾-inch diameter, having the aggregate area of 13·2510 square inches.

The semicircular top of each furnace is extended horizontally to the after smoke connexion, and forms a combustion chamber 7 feet 7 inches in length, and in section a segment 12 inches high of a circle of 28 inches diameter. The water bottom beneath the chambers is 6 inches wide.

The after smoke connexion is in common for all three furnaces; it is a parallelopipedon with the top and bottom connected with the sides by quadrantal curves of 7 inches radius. Its extreme height in the clear is 3 feet 4½ inches, extreme width 18 inches, extreme breadth 8 feet 1 inch. In the back, at the bottom, is an oval manhole with diameters of 12 and 17 inches. The water spaces at the sides are 4 inches wide, at the back 5 inches wide, and at the bottom 6 inches wide, including thicknesses of metal.

From the after smoke connexion the tubes, thirty-four in number, are returned horizontally over the combustion chambers and furnaces to the uptake. Each tube is 6 inches in exterior diameter and 12 feet 6 inches in length; its ends are expanded on one side of the tube plate and riveted over on the other. The space in the clear between the bottom of the lowest row of tubes and the top of the combustion chambers and furnaces is 5½ inches. Thirty of the tubes are distributed in three rows vertically and ten rows horizontally. The remaining four tubes are placed two on each side of the boiler opposite the water spaces between the three vertical rows of tubes. Immediately over the centre of each furnace the horizontal distance between the tubes is 3 inches in the clear. At all other places the distance in the clear between the tubes is 2 inches. The total height occupied by the tubes is 22 inches.

All the tubes discharge into one common uptake at the front of the boiler. Its lower portion is a parallelopipedon with the sides rounded into the bottom on a radius of 7 inches. The top of this portion is drawn at the top of the boiler shell into a circle of 38 inches diameter in the clear, on which the smoke pipe is placed. Breadth in the clear of the lower portion of the uptake 8 feet 1 inch, height 2 feet 1 inch, width in clear of uptake doors 14 inches. The space in the clear between the bottom of the uptake and the crown of the furnaces is 4 inches. The water spaces at the sides of the uptake are 4 inches wide, including thicknesses of metal.

The entire front of the uptake is occupied by three doors closing against narrow cast iron legs. The doors are of wrought iron with a wrought iron lining.

On the top of the boiler shell is placed a cylindrical steam drum of 5 feet 6 inches diameter, and 6 feet height; its top is flat; it communicates with the interior of the shell through a hole of 2 feet diameter. From the side of the drum, near its top, the steam-pipe proceeds to the engine.

The shell has manholes in the spandrels of the furnaces, and in the steam room. It has handholes, also,

in the spandrels of the ash-pits at each end, and in the spandrels of the combustion chambers at the after end.

The flat water spaces of the shell are braced with socket bolts every 9 inches. The top, sides, and ends of the shell are braced every 9 inches, also, with spread braces attached to crow feet.

The bottom of the furnace part of the shell, and the bottoms of the ash-pits are of $\frac{7}{16}$-inch thick plate. The combustion chambers, the after smoke connexion, and the uptake, are of $\frac{5}{16}$-inch thick plate. The tube plates are $\frac{1}{2}$-inch thick. The tubes are lap-welded and of $\frac{1}{8}$-inch thick plate. All other parts are of $\frac{3}{8}$-inch thick plate.

The following are the principal dimensions and proportions of the boiler, namely:—

| | |
|---|---|
| Extreme length, | 15 feet 9 inches. |
| Extreme breadth, | 8 " 9 " |
| Extreme height, exclusive of steam drum, | 8 " 10 " |
| Extreme height, inclusive of steam drum, | 14 " 10 " |
| Number of furnaces, | 3. |
| Breadth of each furnace, | 2 feet 4 inches. |
| Length of each furnace, | 6 " 9 " |
| Total area of grate surface, | 47·25 square feet. |
| Length of combustion chambers, | 7 feet 7 inches. |
| Number of tubes, | 34. |
| Length of each tube, | 12 feet 6 inches. |
| Exterior diameter of each tube, | 6 inches. |
| Interior diameter of each tube, | $5\frac{3}{4}$ " |
| Heating surface in the furnaces, | 108· square feet. |
| Heating surface in the combustion chambers, | 129· " |
| Heating surface in the after smoke connexion, | 73· " |
| Heating surface in the tubes, calculated for their inner circumference, | 639· " |
| Heating surface in the uptake to 12 inches above tubes, | 32· " |
| Total area of water heating surface, | 981· " |
| Total area of steam heating surface in uptake above 12 inches above tubes, | 37· " |
| Diameter of smoke-pipe, | 38 inches. |
| Height of smoke-pipe above grate bars, | 47 feet. |
| Aggregate cross area of combustion chambers for draught, | 6·030 square feet. |
| Aggregate cross area of tubes for draught, | 6·131 " |
| Cross area of the smoke-pipe, | 7·876 " |
| Distance traversed by the products of combustion from the centre of the furnace to their delivery into the uptake, | 28 feet. |
| Capacity of steam room in the boiler, | 238· cubic feet. |
| Capacity of steam room in the steam drum, | 142· " |
| Capacity of steam room in the boiler and steam drum, | 380· " |
| Weight of water in the boiler up to 12 inches above tubes, | 28,750 pounds. |
| Weight of boiler, exclusive of smoke-pipe and grate-bars, but inclusive of all doors and plates, | 49,229 " |

| | |
|---|---|
| Weight of smoke-pipe, | 2,600 pounds. |
| Weight of grate-bars, bearers, &c., | 2,344 " |
| Ratio of the water heating to the grate surface, | 20·762 to 1·000. |
| Ratio of the grate surface to the cross area of the combustion chambers, | 7·836 " |
| Ratio of the grate surface to the cross area of the tubes, | 7·707 " |
| Ratio of the grate surface to the cross area of the smoke-pipe, | 6·000 " |
| Ratio of the steam superheating surface to the grate surface, | 0·783 " |

With the boiler above described and the experiments conducted in the manner narrated, the following results were obtained, namely:—

TABLE CONTAINING THE DATA AND RESULTS OF THE EXPERIMENTS TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER OF THE U. S. STEAMER "YOUNG AMERICA," WITH PENNSYLVANIA ANTHRACITE.

| | | |
|---|---|---|
| | Date of commencing the experiment, | 1 P. M. Oct. 6, 1862. |
| | State of the weather, | Clear with very light breeze. |
| TOTAL QUANTITIES. | Duration of the experiment in consecutive hours, | 72 |
| | Number of cubic feet of water evaporated, | 4,698·98 |
| | Number of pounds of water evaporated, | 292,497·45 |
| | Number of pounds of anthracite consumed, | 38,600· |
| | Number of pounds of refuse from the anthracite in ashes, clinker, and soot, | 6,655· |
| | Number of pounds of combustible consumed, | 31,945· |
| | Per centum of the anthracite in refuse, | 17·24 |
| MEAN QUANTITIES. | Temperature in degrees Fahr. of the external atmosphere in the shade, | 72·0 |
| | Temperature in degrees Fahr. of the fire room, | 91·3 |
| | Temperature in degrees Fahr. of the feed-water in the tank, | 73·2 |
| | Temperature in degrees Fahr. of the products of combustion in the uptake, | 285·6 |
| | Barometer, | 30·06 |
| | Thickness of the bed of anthracite upon the grates, in inches, | 8· |
| | Pounds of anthracite consumed per hour per square foot of grates, | 11·346 |
| | Pounds of combustible consumed per hour per square foot of grates, | 9·390 |
| EVAPORATION. | Total number of pounds of water evaporated from temperature of 100° Fahr., | 299,768·309 |
| | Total number of pounds of water evaporated from temperature of 212° Fahr., | 334,519·379 |
| | Pounds of water evaporated from temperature of 100° Fahr. by one pound of anthracite, | 7·766 |
| | Pounds of water evaporated from temperature of 212° Fahr. by one pound of anthracite, | 8·666 |
| | Pounds of water evaporated from temperature of 100° Fahr. by one pound of combustible, | 9·384 |
| | Pounds of water evaporated from temperature of 212° Fahr. by one pound of combustible, | 10·472 |

# EXPERIMENTS

MADE ON

# THE MACHINE SHOP BOILER

OF THE

## NEW YORK NAVY YARD.

# EXPERIMENTS

MADE ON

# THE MACHINE SHOP BOILER OF THE NEW YORK NAVY YARD,

WITH LOCUST MOUNTAIN AND WITH BLACKHEATH ANTHRACITE TO DETERMINE THEIR RELATIVE EVAPORATIVE EFFICIENCY; AND, ALSO, THE EFFECT PRODUCED UPON THE ECONOMIC EVAPORATION OF THE BOILER BY CONTINUOUSLY DIMINISHING ITS HEATING SURFACE AND CALORIMETER BY STOPPING UP SUCCESSIVE ROWS OF TUBES.

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IN the following pages will be found two series of experiments made, by different experimenters and with anthracites from different localities, upon the Machine Shop Boiler of the New York Navy Yard to determine, first, the evaporative efficiency of the boiler with the particular anthracites burnt and all its tubes in use; and, then, the effect produced upon its economic evaporation by continuously diminishing its heating surface and calorimeter by stopping up successive rows of tubes.

The first series of experiments was conducted by Chief Engineer GARVIN, the Chief Engineer of the New York Navy Yard, and was made with Locust Mountain anthracite taken from the regular deliveries by the coal contractor for the Yard. They embraced four experiments, namely:—the first with all the tubes, consisting of nine rows in height, in use; the second, with the two *upper* rows of tubes stopped off; the third with the three *upper* rows of tubes stopped off; and the fourth with the four *upper* rows of tubes stopped off.

The second series of experiments was conducted by Chief Engineer ZELLER, the Presiding Officer of a Board of Chief Engineers, ordered by the Navy Department to ascertain, by means of the Machine Shop Boiler of the New York Navy Yard, the relative evaporative efficiency of the different coals in the New York and Philadelphia Markets. This series of experiments was made with Blackheath anthracite,

taken, like the Locust Mountain anthracite, from the regular deliveries by the coal contractor for navy steamers. It was more extensive than the other series, and embraced seven experiments, namely:—The first with all the tubes of the boiler in use; the second with the two *lower* rows of tubes stopped off; the third with the three *lower* rows of tubes stopped off; and the fourth with the four *lower* rows of tubes stopped off. Thus far in this as in the other series of experiments the reduction of the heating surface and calorimeter of the boiler had been made by stopping off *horizontal* rows of tubes. This was now varied by effecting the reduction by stopping off *vertical* rows of tubes, the full number of which was eight for each furnace; and in the fifth experiment the two inner vertical rows of each furnace were stopped off. In the sixth experiment the three inner vertical rows were stopped off; and in the seventh experiment the four inner vertical rows of tubes were stopped off.

From the preceding account it will be observed that these experiments determined, for the boiler and coal used, and for the conditions under which they were made, the following facts, namely:—

1st. The evaporative efficiency of the boiler with each kind of anthracite, and all its tubes in use, consisting of nine rows in height and eight rows in width for each of the two furnaces of the boiler.

2d. The effect produced upon the economic evaporation of the boiler by continuously diminishing its heating surface and calorimeter by successively stopping off the two *lower*, the three *lower*, and the four *lower* rows of tubes.

3d. The effect produced upon the economic evaporation of the boiler by continuously diminishing its heating surface and calorimeter by successively stopping off the two *upper*, the three *upper*, and the four *upper* rows of tubes.

4th. The effect produced upon the economic evaporation of the boiler by continuously diminishing its heating surface and calorimeter by successively stopping off the inner two *vertical* rows, the inner three vertical rows, and the inner four vertical rows of tubes of each of the two furnaces.

The effect of continuously reducing the heating surface and calorimeter of the boiler by successively stopping off rows of tubes, was thus determined for the three cases of making this reduction at the top of the mass of tubes, at the bottom of the mass of tubes, and at the side of the mass of tubes for each furnace.

Before proceeding to a particular account of each series of experiments, the following description of the boiler is given, in order that the whole of the conditions may be completely understood.

## BOILER.

The boiler is of the horizontal tubular type with the tubes returned above the furnaces. It was new and had been in use but a few weeks. Its shell was double riveted, and perfect tightness had been assured for a working pressure of 60 pounds per square inch above the atmosphere. It was covered over its entire exterior with a thick coat of new felt stitched to thick canvass.

The shell is 12 feet long, 7 feet 6 inches wide, and 12 feet high. The top is semicircular, and the bottom and sides are flat. It has neither steam drum nor steam chimney.

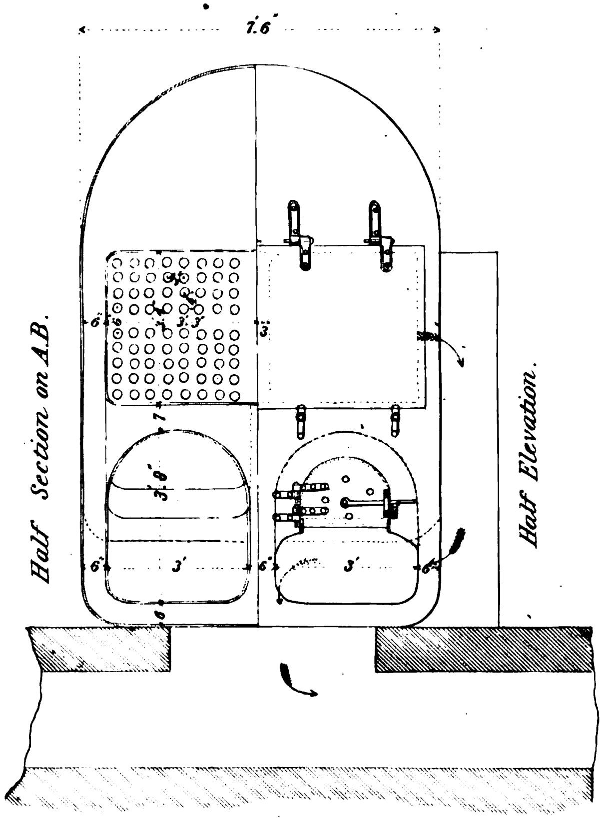
The furnaces are two in number and semicircular on top. They are 6 feet long and 3 feet wide. Their crown is 22 inches above the grate bars at front and 26 inches above them at back.

The door opening for each furnace is semicircular on top; it is 15 inches wide and 16 inches in extreme height. In each furnace door there are four holes of 1¼ inches diameter, making an aggregate area of [illegible] square inches; the [illegible] plate of the furnace door is perforated with one hundred holes of [illegible]-inch diameter.

The total height from the bottom of the ash-pit to the crown of the furnace is 44 inches, leaving the

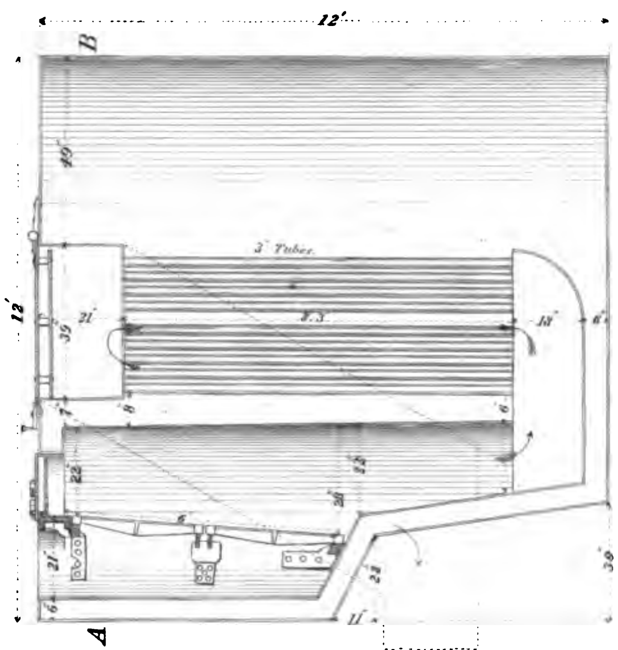
# BOILER OF THE MACHINE SHOP ENGINE,

*U.S.NAVY-YARD, NEW-YORK.*




*Half Section on A.B.*

*Half Elevation.*




*Section through centre of Furnace and Shell.*

height from the bottom of the ash-pit to the top of the grate-bars 22 inches at the front and 16 inches at the back. The angles of the ash-pits are rounded on a radius of 6 inches.

From each furnace a combustion chamber 42 inches long extends to the back smoke connexion. This chamber is, on top, a continuation of the crown of the furnace; on the bottom it is flat, with the angles rounded on a radius of 4 inches.

The back smoke connexion is in two separate compartments, one for each furnace, so that all the products of combustion from each furnace are compelled to pass through the tubes immediately over that furnace. This connexion is 18 inches wide in the clear and 62 inches in extreme height.

The boiler contains one hundred and forty-four tubes. They are of iron 3 inches in external diameter, $2\frac{3}{4}$ inches in internal diameter, and 8 feet 3 inches in extreme length. The least space between them horizontally in the clear is $1\frac{1}{2}$ inch and vertically 1 inch. They are secured in their plates by being expanded on one side and riveted over on the other. They are nine rows in height, and occupy a space vertically of 37 inches. The distance in the clear between the crown of the furnace and combustion chamber, and the lowest tube is 8 inches at front of boiler and 6 inches at back of boiler.

The front smoke connexion or uptake is 21 inches wide and extends across the boiler. After leaving the front smoke connexion the products of combustion are carried through a sheet iron flue along one side of the boiler in a direction sloping downwards and delivered into a horizontal flue of masonry in the floor beneath the boiler, whence they are conveyed to the chimney.

All the flat water spaces are 6 inches wide.

The grate bars are in two lengths; they are 1 inch wide on top and have $\frac{5}{8}$-inch air spaces between them.

A manhole is placed in each spandrel of the furnace arches, and a handhole is placed in each spandrel at the ash-pit corners. The smoke-box doors are lined and accurately fitted to the shell.

The bottoms of the ash-pits and of the front of the boiler are of $\frac{7}{16}$-inch thick plate; the tube plates are $\frac{1}{2}$-inch thick; the tubes are $\frac{1}{10}$-inch thick; all other parts of the boiler are of $\frac{3}{8}$-inch thick plate. All seams not in contact with the fire are double riveted.

The following are the principal dimensions and proportions of the boiler required to be known, namely:—

| | |
|---|---|
| Extreme length of boiler, | 12 feet. |
| Extreme breadth of boiler, | 7 " 6 inches. |
| Extreme height of boiler, | 12 " |
| Number of furnaces in boiler, | 2 |
| Length of each furnace, | 6 feet. |
| Breadth of each furnace, | 3 " |
| Area of grate surface in the two furnaces, | 36 square feet. |
| Total number of tubes, | 144 |
| Length of each tube, | 8 feet 3 inches. |
| External diameter of each tube, | 3 inches. |
| Internal diameter of each tube, | $2\frac{4}{5}$ " |
| Area of heating surface in furnaces, | 79·48 square feet. |
| Area of heating surface in combustion chambers, | 33·32 " |
| Area of heating surface in back smoke connexion, | 78·03 " |

| | |
|---|---|
| Area of heating surface in tubes (calculated for their inside diameter), | 862·40 square feet. |
| Area of heating surface in uptake to 9 inches above top of tubes, . | 61·33 " |
| Area of heating surface in side of boiler in sheet iron flues, . . . | 29·04 " |
| Total area of heating surface, . . . . . . | 1,143 60 " |
| Area of water level, . . . . . . . | 90· " |
| Capacity of steam room in shell, . . . . . | 226· cubic feet. |
| Capacity of water space up to 9 inches above tubes, . . . | 370· " |
| Height of smoke-pipe above grate-bars, . . . . . | 72 feet. |
| Cross area of the two combustion chambers, . . . . | 7·068 square feet. |
| Cross area of all the tubes (calculated for their inside diameter), . | 6·157 " |
| Ratio of the heating to the grate surface, . . . . . | 31·767 to 1·000. |
| Ratio of the grate surface to the cross area of the combustion chambers, | 5·093 " |
| Ratio of the grate surface to the cross area of the tubes, . . | 5·847 " |
| Distance traversed by the products of combustion from the centre of the furnace to their emergence from the tubes, . . . | 18¾ feet. |
| Weight of boiler, exclusive of grate bars, but inclusive of doors, handhole and manhole plates, . . . . . . | 31,500 pounds. |
| Weight of water in boiler, at 62·321 pounds to the cubic foot, . | 23,059 " |

In the following table will be found such of the dimensions and proportions of the boiler, both as originally constructed and as modified for the following experiments by stopping up the rows of tubes previously described, as are required for the clear comprehension of the changes thereby produced in the heating surface and in the calorimeter, the effects of which were the results to be ascertained.

## TABLE SHOWING THE PRINCIPAL DIMENSIONS AND PROPORTIONS OF THE BOILER AS ORIGINALLY CONSTRUCTED, AND AS MODIFIED FOR THE FOLLOWING EXPERIMENTS BY STOPPING UP CERTAIN ROWS OF TUBES BOTH HORIZONTAL AND VERTICAL.

| | As originally constructed. | As modified by having the heating surface and calorimeter successively diminished by stopping up horizontal rows of tubes. | | | As modified by having the heating surface and calorimeter successively diminished by stopping up vertical rows of tubes. | | |
|---|---|---|---|---|---|---|---|
| | | Two horizontal rows of tubes of each furnace stopped at both ends. | Three horizontal rows of tubes of each furnace stopped at both ends. | Four horizontal rows of tubes of each furnace stopped at both ends. | Two vertical rows of tubes of each furnace stopped at both ends. | Three vertical rows of tubes of each furnace stopped at both ends. | Four vertical rows of tubes of each furnace stopped at both ends. |
| Area of grate surface, in square feet, . . | 36· | 36· | 36· | 36· | 36· | 36· | 36· |
| Area, in square feet, of heating surface in furnaces, . | 79·48 | 79·48 | 79·48 | 79·48 | 79·48 | 79·48 | 79·48 |
| Area, in square feet, of heating surface in combustion chambers, . . . . . . | 33·32 | 33·32 | 33·32 | 33·32 | 33·32 | 33·32 | 33·32 |
| Area, in square feet, of heating surface in back smoke connexions, . . . . . . | 78·03 | 78·03 | 78·03 | 78·03 | 78·03 | 78·03 | 78·03 |
| Area, in square feet, of heating surface in tubes (calculated for inside diameter), . . . . | 862·40 | 670·75 | 574·93 | 479·11 | 646·80 | 539·00 | 481·20 |
| Area, in square feet, of heating surface in uptake (to 9 ins. above top of tubes), . . . . . | 61·33 | 61·33 | 61·33 | 61·33 | 61·33 | 61·33 | 61·33 |
| Area, in square feet, of heating surface in side of boiler in sheet iron flue, . . . . . | 29·04 | 29·04 | 29·04 | 29·04 | 29·04 | 29·04 | 29·04 |
| Total area of all heating surface, in square feet, . | 1143·60 | 951·95 | 856·13 | 760·31 | 928·00 | 820·20 | 712·40 |
| Cross area of the tubes, in square feet, (calculated for inside diameter,) . . . . . | 6·157 | 4·789 | 4·104 | 3·420 | 4·618 | 3·848 | 3·078 |
| Number of square feet of heating surface per square foot of grate surface, . . . . . | 31·767 | 26·448 | 23·781 | 21·120 | 25·778 | 22·783 | 19·789 |
| Number of square feet of grate surface per square foot of cross area of tubes, . . . | 5·847 | 7·517 | 8·772 | 10·526 | 7·796 | 9·355 | 11·696 |

## EXPERIMENTS MADE WITH LOCUST MOUNTAIN ANTHRACITE ON THE MACHINE SHOP BOILER OF THE NEW YORK NAVY YARD, TO DETERMINE ITS EVAPORATIVE EFFICIENCY WITH THAT COAL, AND THE EFFECT UPON THE ECONOMIC EVAPORATION OF DIMINISHING THE HEATING SURFACE AND CALORIMETER BY STOPPING UP SUCCESSIVELY THE TWO UPPER, THE THREE UPPER, AND THE FOUR UPPER HORIZONTAL ROWS OF TUBES OF EACH OF THE TWO FURNACES.

In the following experiments on the boiler of the machine shop of the New York Navy Yard, Locust Mountain anthracite was employed. The object of the experiments was to determine; 1st, The evaporative efficiency of the boiler with this anthracite and, as originally constructed, with all the tubes in use. 2d, The effect produced upon the economic evaporation by stopping up, in succession, the two upper, the three upper, and the four upper horizontal rows of tubes. The stopping up was done by closing both ends of the tubes with tight fitting cast iron plugs.

The coal was all taken from the same heap, and in the condition delivered by the contractor for the regular supplies of the Yard. It was in uniform lumps of moderate size and free from dust and dirt. The per centum of refuse varied from 16·83 to 18·06 in the different experiments.

The experiments were not of equal duration; neither were they made in the order in which they stand in the following table. The experiment *with all the tubes in use,* recorded in column A, continued ninety-six consecutive hours; and was followed by the experiment *with the three upper rows of tubes stopped,* recorded in column C, and which continued fifty and one-third hours. This was immediately followed by the experiment *with the two upper rows of tubes stopped,* recorded in column B, and continuing twenty-four hours. It would have been continued forty-eight hours had the boiler not been unexpectedly wanted for pumping out the dry dock. The last experiment recorded in column D, was made after the lapse of eight days and *with the four upper rows of tubes stopped;* it continued forty-eight hours.

The total number of rows of tubes, vertically, was nine; consequently, in experiment B, the tube surface and the calorimeter were reduced to seven-ninths of what they were in experiment A. In experiment C they were reduced to two-thirds or six-ninths: and in experiment D to five-ninths. The *total* heating surface of the boiler, however, was, by these stoppings off, only reduced successively to 83·24, 74·87, and 58·10 per centum of the area with all these tubes in use; or in round fractions to $\frac{10}{12}$ths, $\frac{9}{12}$ths, and $\frac{7}{12}$ths.

## MANNER OF MAKING THE EXPERIMENTS.

The experiments were conducted by four Assistant Engineers of the Navy under the supervision of Chief Engineer GARVIN. The same firemen, instruments, and tools were employed throughout; and the greatest care taken that the conditions should be uniform throughout each and all of the experiments, and the results rigorously exact.

The experiments were made in precisely the same manner and with the same instruments, as hereinafter described under the head of those made for similar purpose on the same boiler with Blackheath anthracite by the Board ordered by the Navy Department to determine the relative economic evaporative efficiency of the various kinds of coals in the markets of New York and Philadelphia. The data and results will be found in Table No. 1, as follows, namely:—

## TABLE No. 1:—CONT

ANTHRACITE ON THE
UPON THE ECONOMIC
PING UP SUCCESSIVEI

| |
|---|
| Date of commencing the experiment, |
| Duration of the experiment, in consecutive |
| State of the weather, . . |
| Cubic feet of water evaporated, as measur |
| Pounds of water evaporated from the temp |
| Pounds of Locust Mountain anthracite co |
| Pounds of refuse from the anthracite, in |
| Pounds of combustible consumed, |
| Per centum of the anthracite, in ashes, c |
| Pounds of anthracite consumed per hour |
| Pounds of combustible consumed per hou |
| Thickness of the bed of anthracite upon |
| Temperature in degrees Fahr. of the ext |
| Temperature in degrees Fahr. of the fir |
| Temperature in degrees Fahr. of the wa |
| Temperature in degrees Fahr. of the pr uptake, . . |
| Barometer, . . . |
| Pounds of water evaporated from the t |
| Pounds of water evaporated from the t |
| Pounds of water evaporated by one pou ture of 100° Fahr., . |
| Pounds of water evaporated by one po ture of 212° Fahr., . |
| Pounds of water evaporated by one po rature of 100° Fahr., . |
| Pounds of water evaporated by one po rature of 212° Fahr., . |
| Comparative evaporative efficiency of |
| Comparative evaporative efficiency of |

taken, like the Locust Mountain anthracite, from the regular deliveries by the coal contractor for naval steamers. It was more extensive than the other series, and embraced seven experiments, namely:—The first, with all the tubes of the boiler in use; the second, with the two *lower* rows of tubes stopped off; the third, with the three *lower* rows of tubes stopped off; and the fourth with the four *lower* rows of tubes stopped off. Thus far in this, as in the other series of experiments, the reduction of the heating surface and calorimeter of the boiler had been made by stopping off *horizontal* rows of tubes. This was now varied by effecting the reduction by stopping off *vertical* rows of tubes, the full number of which was eight for each furnace, and in the fifth experiment the two inner vertical rows of each furnace were stopped off. In the sixth experiment, the three inner vertical rows were stopped off; and in the seventh experiment, the four inner vertical rows of tubes were stopped off.

From the preceding account it will be observed that these experiments determined for the boiler and coal used, and for the conditions under which they were made, the following facts, namely:—

1st. The evaporative efficiency of the boiler with each kind of anthracite, and all its tubes in use, consisting of nine rows in height and eight rows in width for each of the two furnaces of the boiler.

2d. The effect produced upon the economic evaporation of the boiler by continuously diminishing its heating surface and calorimeter by successively stopping off the two *lower*, the three *lower*, and the four *lower* rows of tubes.

3d. The effect produced upon the economic evaporation of the boiler by continuously diminishing its heating surface and calorimeter by successively stopping off the two *upper*, the three *upper*, and the four *upper* rows of tubes.

4th. The effect produced upon the economic evaporation of the boiler by continuously diminishing its heating surface and calorimeter by successively stopping off the inner two *vertical* rows, the inner three vertical rows and the inner four vertical rows of tubes of each of the two furnaces.

The effect of continuously reducing the heating surface and calorimeter of the boiler by successively stopping of rows of tubes, was thus determined for the three cases of making this reduction at the top of the mass of tubes, at the bottom of the mass of tubes, and at the side of the mass of tubes for each furnace.

Before proceeding to a particular account of each series of experiments, the following description of the boiler is given in order that the whole of the conditions may be completely understood.

## BOILER.

The boiler is of the horizontal tubular type with the tubes returned above the furnaces. It was new, and had been in use but a few weeks. Its shell was double riveted, and perfect tightness had been assured for a working pressure of 60 pounds per square inch above the atmosphere. It was covered over its entire exterior with a thick coat of new felt stitched to thick canvass.

The shell is 12 feet long, 7 feet 6 inches wide, and 12 feet high. The top is semicircular, and the bottom and sides are flat. It has neither steam drum nor steam chimney.

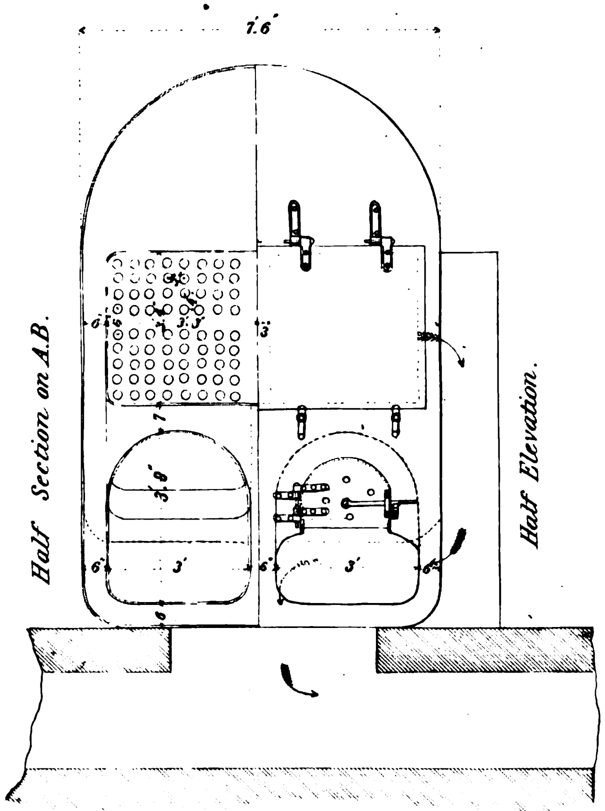
The furnaces are two in number and semicircular on top. They are 6 feet long, and 3 feet wide. Their crown is 22 inches above the grate bars at front and 28 inches above them at back.

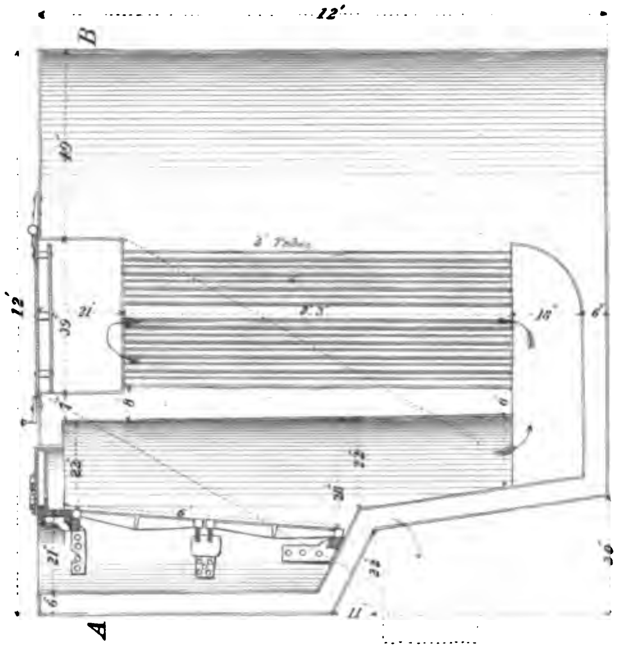
The door opening for each furnace is semicircular on top, it is 18 inches wide and 16 inches in extreme height. In each furnace door there are four holes of $1\frac{3}{4}$ inches diameter, making an aggregate area of 9·62 square inches; the lining plate of the furnace door is perforated with one hundred holes of $\frac{3}{8}$-inch diameter.

The total height from the bottom of the ash-pit to the crown of the furnace is 44 inches, leaving the

# BOILER OF THE MACHINE SHOP ENGINE,

## U.S.NAVY-YARD, NEW-YORK.




*Section through centre of Furnace and Shell.*

height from the bottom of the ash-pit to the top of the grate-bars 22 inches at the front and 16 inches at the back. The angles of the ash-pits are rounded on a radius of 6 inches.

From each furnace a combustion chamber 42 inches long extends to the back smoke connexion. This chamber is, on top, a continuation of the crown of the furnace; on the bottom it is flat, with the angles rounded on a radius of 4 inches.

The back smoke connexion is in two separate compartments, one for each furnace, so that all the products of combustion from each furnace are compelled to pass through the tubes immediately over that furnace. This connexion is 18 inches wide in the clear and 62 inches in extreme height.

The boiler contains one hundred and forty-four tubes. They are of iron 3 inches in external diameter, $2\frac{5}{8}$ inches in internal diameter, and 8 feet 3 inches in extreme length. The least space between them horizontally in the clear is $1\frac{1}{8}$ inch and vertically 1 inch. They are secured in their plates by being expanded on one side and riveted over on the other. They are nine rows in height, and occupy a space vertically of 37 inches. The distance in the clear between the crown of the furnace and combustion chamber, and the lowest tube is 8 inches at front of boiler and 6 inches at back of boiler.

The front smoke connexion or uptake is 21 inches wide and extends across the boiler. After leaving the front smoke connexion the products of combustion are carried through a sheet iron flue along one side of the boiler in a direction sloping downwards and delivered into a horizontal flue of masonry in the floor beneath the boiler, whence they are conveyed to the chimney.

All the flat water spaces are 6 inches wide.

The grate bars are in two lengths; they are 1 inch wide on top and have $\frac{5}{8}$-inch air spaces between them.

A manhole is placed in each spandrel of the furnace arches, and a handhole is placed in each spandrel at the ash-pit corners. The smoke-box doors are lined and accurately fitted to the shell.

The bottoms of the ash-pits and of the front of the boiler are of $\frac{7}{16}$-inch thick plate; the tube plates are $\frac{1}{2}$-inch thick; the tubes are $\frac{1}{16}$-inch thick; all other parts of the boiler are of $\frac{3}{8}$-inch thick plate. All seams not in contact with the fire are double riveted.

The following are the principal dimensions and proportions of the boiler required to be known, namely:—

| | |
|---|---|
| Extreme length of boiler, | 12 feet. |
| Extreme breadth of boiler, | 7 " 6 inches. |
| Extreme height of boiler, | 12 " |
| Number of furnaces in boiler, | 2 |
| Length of each furnace, | 6 feet. |
| Breadth of each furnace, | 3 " |
| Area of grate surface in the two furnaces, | 36 square feet. |
| Total number of tubes, | 144 |
| Length of each tube, | 8 feet 3 inches. |
| External diameter of each tube, | 3 inches. |
| Internal diameter of each tube, | $2\frac{5}{8}$ " |
| Area of heating surface in furnaces, | 79·48 square feet. |
| Area of heating surface in combustion chambers, | 33·32 " |
| Area of heating surface in back smoke connexion, | 78·03 " |

The temperature of the feed-water in the tank was denoted by a thermometer kept constantly suspended in it. The temperature of the products of combustion in the uptake was ascertained by means of a mercurial thermometer immersed in oil contained in a copper pot of 3 inches diameter and 12 inches length. This pot was suspended in the uptake close to and opposite the centre of the mass of tubes belonging to the furnace adjacent that side of the boiler whence the gases left it. The pot could be taken out and the thermometer read without removing it from the oil before there was any appreciable fall in its temperature.

Each experiment,—with the exception of the one with all the tubes in use, recorded in column A, and which was continued for 72 continued consecutive hours,—lasted 48 consecutive hours. The evaporation was performed under the atmospheric pressure, and the steam escaped through a temporary steam escape-pipe fitted to an oval manhole with diameters of 16 and 18 inches. The manhole was at the centre of the top of the shell of the boiler; the pipe was of equal area; and was elbowed down immediately after leaving the manhole, and pierced at the lowest point, so as by constant drainage to prevent any of the water resulting from the condensation of steam in the pipe from returning to the boiler.

Before commencing an experiment the tubes were swept, and the uptake, back connexions, furnaces, &c., thoroughly cleaned. The water was first brought to the boiling point by wood, which was then allowed to burn down to the few live coals required to kindle the anthracite. The water level was now adjusted to precisely 9 inches above the top of the tubes, the time noted, the anthracite fired, and the experiment held to commence. At the end of the experiment the anthracite was allowed to burn as nearly out as possible, and at the exact expiration of the 48 hours, the water level having been scrupulously brought to the same mark as at the commencement, the furnaces were drawn and the unconsumed coal picked out from the refuse, weighed, and deducted from the total quantity expended. The sweepings of the tubes, uptake, &c., were also weighed dry and added to the amount of refuse.

The experiments were carried on, under the direction of the Board, by four experienced assistant engineers, who stood regular watches of six hours each, and personally filled the tank, weighed the coal and ashes, observed the temperatures, directed the firing, &c., and noted all the data of the experiments. During each experiment a tabular record was kept, in which were hourly entered the temperature of the external atmosphere in the shade, of the fire room, of the water in the tank, and of the products of combustion in the uptake. Also, the height of the barometer, and the number of pounds of coal thrown into the furnaces during the hour. In appropriate columns were entered the weight of refuse and the condition of the weather during the watch. The exact time each tankful of water was emptied was recorded in a separate column, and served as a check upon error in this important quantity, as, owing to the uniformity in the rate of evaporation, the omission or undue addition of a tankful could not escape detection in the final scrutiny. The exact time of commencing and ending each experiment was entered in its record, as well as the weight of refuse and of unconsumed coal withdrawn from the furnaces at its close. The sweepings of the tubes, &c., were also separately noted.

Each experiment was commenced and closed by Chief Engineer Zeller, the presiding engineer officer of the Board, who also supervised it from time to time during its progress, and saw that the instructions for its conduct were observed with the minutest accuracy. It is believed the results may be relied on with perfect confidence: they will be found, together with the data, in the following table, namely:—

## ADDITIONAL EXPERIMENTS MADE WITH AN ANTHRACITE FROM THE WESTERN PORTION OF THE PENNSYLVANIA MIDDLE COAL FIELD TO DETERMINE THE DIFFERENCE IN THE EVAPORATIVE EFFICIENCY OF THE BOILER OF THE MACHINE SHOP OF THE U. S. NAVY YARD, NEW YORK, WHEN ALL THE TUBES ARE IN USE, AND WHEN THE LOWER TWO ROWS OF TUBES ARE STOPPED.

Advantage was taken of a convenient opportunity which occurred some time after the date of the preceding first set of experiments with Locust Mountain anthracite, and previous to the preceding second set of experiments with Blackheath anthracite, during the course of trials of the comparative evaporative efficiencies of different coals, to ascertain the effect with a still different anthracite upon the economic evaporation of the machine shop boiler by stopping the lower two rows of tubes.

The coal used in this experiment was an anthracite of uncertain precise locality, but from Schuylkill County, near the western portion of the middle coal field of Pennsylvania. It was softer than the Pennsylvania anthracites known in the market, and made, in burning, a vitreous clinker which adhered strongly to the grate-bars, filled up their interstices, and required much time and labor for removal. The greater length of time the furnace doors were open in cleaning fires with this coal must have somewhat affected its evaporative efficiency.

The experiments were conducted in the same manner as the preceding ones, and by the Board ordered by the Navy Department to determine the comparative economic evaporation of different coals. The data and results will be found in the following table, in which the first column contains the record of an experiment of seventy-two consecutive hours duration *with all the tubes in use:* and the last column the record of one of forty-eight consecutive hours duration *with the lower two rows of tubes stopped at both ends* by closely fitting cast iron plugs.

**TABLE No. 3,** CONTAINING THE DATA AND RESULTS OF EXPERIMENTS MADE WITH THE MACHINE SHOP BOILER OF THE NEW YORK NAVY YARD, TO ASCERTAIN THE EFFECT UPON ITS ECONOMIC EVAPORATION OF STOPPING ITS LOWER TWO ROWS OF TUBES.

| | | ALL THE TUBES IN USE. | THE LOWER TWO ROWS OF TUBES STOPPED. |
|---|---|---|---|
| | Date of commencing the experiment, | 9½ A.M. July 21, 1862. | 12½ P.M. July 25, 1862. |
| | Date of ending the experiment, | 9½ A.M. July 24, 1862. | 12½ P.M. July 27, 1862. |
| | State of the weather, | Two-thirds of the time moderate breezes and cloudy. One-third of the time moderate breezes and raining. | Gentle breezes and mostly clear sky. |
| TOTAL QUANTITIES. | Duration of the experiment in consecutive hours, | 72· | 48· |
| | Number of cubic feet of water evaporated of the temperature of the tank, | 3,926·448 | 2,458·744 |
| | Number of pounds of water evaporated, | 244,471·568 | 153,091·795 |
| | Number of pounds of anthracite consumed, | 31,868· | 20,656· |
| | Number of pounds of refuse from the anthracite, in ashes, &c., | 4,514· | 2,882· |
| | Number of pounds of combustible consumed, | 27,354· | 17,774· |
| | Per centum of the anthracite in refuse, | 14·16 | 13·95 |
| MEAN QUANTITIES. | Temperature in degrees Fahr. of the external atmosphere in the shade, | 69·2 | 78·7 |
| | Temperature in degrees Fahr. of the fire room, | 87·1 | 90·0 |
| | Temperature in degrees Fahr. of the water in the tank, | 71·2 | 71·0 |
| | Temperature in degrees Fahr. of the products of combustion in the uptake, | 373·7 | 380·8 |
| | Barometer, | 30·01 | 29·97 |
| | Thickness of the bed of anthracite upon the grates, in inches, | 7· | 7· |
| | Pounds of anthracite consumed per hour per square foot of grate surface, | 12·295 | 11·954 |
| | Pounds of combustible consumed per hour per square foot of grate surface, | 10·553 | 10·286 |
| EVAPORATION. | Total number of pounds of water evaporated from temperature of 100° Fahr., | 251,002·299 | 157,210·219 |
| | Total number of pounds of water evaporated from temperature of 212° Fahr., | 280,101·003 | 175,487·491 |
| | Pounds of water evaporated from temperature of 100° Fahr. by one pound of anthracite, | 7·876 | 7·611 |
| | Pounds of water evaporated from temperature of 212° Fahr. by one pound of anthracite, | 8·789 | 8·493 |
| | Pounds of water evaporated from temperature of 100° Fahr. by one pound of combustible, | 9·176 | 8·845 |
| | Pounds of water evaporated from temperature of 212° Fahr. by one pound of combustible, | 10·240 | 9·870 |
| | Comparative evaporative efficiency of the pound of anthracite, | 1·000 | 0·966 |
| | Comparative evaporative efficiency of the pound of combustible, | 1·000 | 0·964 |

## DISCUSSION OF THE RESULTS OF THE PRECEDING EXPERIMENTS.

The preceding experiments determine, for the boiler employed and the conditions under which they were made; 1st. The comparative economic evaporative efficiency of the three kinds of anthracite consumed.

2d. The effect produced upon the economic evaporative efficiency by stopping up successively the *two upper*, the *three upper*, and *the four upper horizontal rows of tubes.*

3d. The effect produced upon the economic evaporative efficiency by stopping up successively the *two lower* the *three lower*, and the *four lower horizontal rows of tubes.*

4th. The effect produced upon the economic evaporative efficiency by stopping up successively the *inner two vertical rows*, the *inner three vertical rows*, and the *inner four vertical rows of tubes* of each of the two furnaces of the boiler.

In making the comparisons the economic effect will be measured by the number of pounds of water evaporated from the temperature of 100° Fahr. by one pound of combustible. It is here to be understood that by the term "combustible" is meant the remainder of the anthracite after deducting the refuse from it in ashes, clinker, soot, and dust. That the weight of combustible and not the weight of anthracite should be taken as the true measure of the cost of evaporation, is obvious from the fact that the per centum of refuse is an accidental proportion and variable in every shovelful of the same coal.

And 1st. *Of the comparative economic evaporative efficiency of the three kinds of anthracite employed.* Making the comparison by the results when all the tubes were in use, we have the following number of pounds of water evaporated from the temperature of 100° Fahr., per pound of combustible, namely:—

| | | |
|---|---|---|
| Locust Mountain anthracite, | 9·070 or | 1·000 |
| Blackheath anthracite, | 10·112 " | 1·115 |
| Anthracite of unknown name, | 9·176 " | 1·012 |

An experiment of 72 consecutive hours duration, made on the same Locust Mountain anthracite, but from a different cargo, by the Board of which Chief Engineer ZELLER was the presiding officer, gave an evaporation from 100° Fahr. of 9·695 pounds of water per pound of combustible, the per centum of refuse being 19·72. The two different specimens of the same anthracite consequently had evaporative efficiencies in the ratio of $\left(\frac{9{\cdot}695}{9{\cdot}070}=\right)$ 1·000 to 1·069.

2d. *Of the effect produced upon the economic evaporative efficiency by stopping successively the two upper, the three upper, and the four upper horizontal rows of tubes.* The comparative economic evaporative efficiency, taken from Table No. 1, containing the data and results of the experiments with the Locust Mountain anthracite, stands as follows, namely:—

| | |
|---|---|
| With all the tubes in use, | 1·000 |
| With the two upper horizontal rows stopped, | 1·069 |
| With the three upper horizontal rows stopped, | 1·168 |
| With the four upper horizontal rows stopped, | 1·192 |

In this set of experiments the economic evaporative efficiency of the anthracite increases regularly and in a high proportion as the heating surface and calorimeter of the boiler diminishes. The difference is too

great and in too uniform a ratio to be the result of any insensible difference in the mechanical conditions of the anthracite or of its firing; and the experiments were too carefully conducted to admit of error in the quantities. We are, therefore, compelled to allow that up to the suppression of the four upper horizontal rows of tubes, out of nine rows, the economic evaporative efficiency increased in a surprising degree. The total areas of heating surface, and the rates of combustion, which latter did not vary in the direction to favorably influence the economic result, were, comparing the number of pounds of combustible consumed per hour per square foot of grate surface, as follows, namely:—

| | Rates of Combustion. | Areas of Heating Surface. |
|---|---|---|
| With all the tubes in use, | 1·000 | 100·00 |
| With the two upper horizontal rows stopped, | 1·142 | 83·24 |
| With the three upper horizontal rows stopped, | 0·840 | 74·87 |
| With the four upper horizontal rows stopped, | 0·891 | 66·48 |

It will be observed that with each suppression of tube surface and calorimeter the temperature of the products of combustion in the boiler uptake regularly increased, with the exception of the experiment, (Table No. 1, column B) with the two upper rows stopped in which the disproportionately high temperature may have been due to the high rate of combustion.

3d. *Of the effect produced upon the economic evaporative efficiency by stopping up successively the two lower, the three lower, and the four lower horizontal rows of tubes.* The comparative economic evaporative efficiency, taken from Table No. 2, columns A, B, C, and D, containing the data and results of the experiments made with Blackheath anthracite, stand as follows, namely:—

| | |
|---|---|
| With all the tubes in use, | 1·000 |
| With the two lower horizontal rows stopped, | 0·924 |
| With the three lower horizontal rows stopped, | 1·000 |
| With the four lower horizontal rows stopped, | 1·030 |

This set of experiments was precisely like that of the preceding one (Table No. 1) with the simple exception that the rows of tubes stopped up were at the bottom of the mass of tubes instead of at the top. This difference caused the tubes in use to be nearer to the surface of the water and consequently to have a less hydrostatic pressure upon them. They were likewise further removed from the crown of the furnaces; and by reason of the greater space thus left between the lower row of tubes in use and the crown of the furnaces, the steam generated on the surfaces of the latter may have had time and space to pass off obliquely to the right and left, and ascend to the surface of the water through the wide spaces between the masses of tubes over each furnace, and between these masses and the sides of the boiler, instead of ascending between the tubes. The result would be a less rapid circulation of the water between the tubes with the lower rows stopped than with the upper rows stopped. Be this as it may, the difference in the effects is very great. Instead of the strongly marked increase in the economic evaporative efficiency, which in the former set of experiments (Table No. 1) accompanied the suppression of each row of tubes, we have a decrease of 7·6 per centum with two rows stopped, equality with three rows stopped, and a gain of three per centum with four rows stopped. If now, we refer to Table No 3 we shall find that the suppression of the two lower rows of tubes was followed by a diminution of 3·6 per centum in the economic evaporative efficiency, which is in the same direction though less in degree as the above. From this set of experiments we are entitled to conclude that the economic evaporative efficiency was not diminished by the suppression of the lower horizontal rows of tubes up to four rows out of nine.

The total areas of heating surface in use and the rates of combustion, making the comparison for the latter by the number of pounds of combustible consumed per hour per square foot of grate surface, were as follows, namely:—

| | Rates of Combustion. | Areas of Heating Surfaces. |
|---|---|---|
| With all the tubes in use, . . . | 1·000 | 100·00. |
| With the two lower horizontal rows stopped, . | 0·868 | 83·24. |
| With the three lower horizontal rows stopped, | 0·955 | 74·87. |
| With the four lower horizontal rows stopped, . | 0·923 | 66·48. |

With this set of experiments (Table No. 2, columns A, B, C, and D), as with the preceding one, (Table No. 1,) it will be observed that, at each suppression of tube surface and calorimeter, the temperature of the products of combustion in the boiler uptake regularly increased, with the exception of the experiment (Table No. 2, column B) with the two lower rows stopped, in which this temperature was slightly lower than in the experiment (Table No. 2, column A) with all the tubes in use.

4th. *Of the effect produced upon the economic evaporative efficiency by stopping up successively the inner two vertical rows, the inner three vertical rows, and the inner four vertical rows of tubes of each of the two furnaces.* The comparative economic evaporative efficiency, taken from Table No. 2, columns A, E, F, and G, containing the data and results of the experiments made with Blackheath anthracite, stand as follows, namely:—

| | |
|---|---|
| With all the tubes in use, . . . . . . | 1·000. |
| With the inner two vertical rows of tubes of each furnace stopped, . . | 0·930. |
| With the inner three vertical rows of tubes of each furnace stopped, . . | 0·965. |
| With the inner four vertical rows of tubes of each furnace stopped, . . | 0·990. |

This set of experiments (Table No. 2, columns A, E, F, and G) varied from the preceding ones (Table No. 1 and Table No. 2, columns A, B, C, and D), in that the rows of tubes suppressed were vertical instead of horizontal, and the tube surface and calorimeter remaining after each suppression was $\frac{6}{8}$ths instead of $\frac{7}{9}$ths; $\frac{5}{8}$ths instead of $\frac{6}{9}$ths: and $\frac{4}{8}$ths instead of $\frac{5}{9}$ths. By making the suppressions on the inner rows of each of the two furnaces of the boiler, a very wide space was left at the centre of the boiler between the masses of tubes in use. This allowed the principal portion of the steam generated by the furnaces to pass up this wide space, and the spaces between the masses of tubes and the sides of the boiler, instead of between the tubes, thereby lessening the water circulation among the latter.

The total areas of heating surface in use and the rates of combustion, making the comparison for the latter by the number of pounds of combustible consumed per hour per square foot of grate surface, were as follows, namely:—

| | Rates of Combustion. | Areas of Heating Surface. |
|---|---|---|
| With all the tubes in use, . . . | 1·000 | 100·00. |
| With the inner two vertical rows of tubes of each furnace stopped, | 0·874 | 81·15. |
| With the inner three vertical rows of tubes of each furnace stopped, | 0·990 | 71·72. |
| With the inner four vertical rows of tubes of each furnace stopped, | 0·951 | 62·29. |

The results from this set of experiments, though 7 per centum less than with all the tubes in use when the two inner vertical rows of tubes of each furnace were suppressed, showed a gradual increase in economic evaporative efficiency from that experiment (Table No. 2, column E) with each succeeding suppression

of a vertical row of tubes up to the suppression of four rows (Table No. 2, column G), when the economic evaporative efficiency was very nearly equal to that with all the tubes in use (Table No. 2, column A).

In this set of experiments the temperature of the products of combustion in the boiler uptake regularly increased with each suppression of tube surface and calorimeter.

Let us now compare the results from all the preceding experiments by grouping them in the following table, namely:—

| | All the tubes in use. | | | Two rows of tubes suppressed. | | | Three rows of tubes suppressed. | | | Four rows of tubes suppressed. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Pounds of water evaporated from 100° Fahr. by one pound of combustible. | Pounds of combustible consumed per hour per square foot of grates. | Temperature in degrees Fahr. of the products of combustion in the uptake. | Pounds of water evaporated from 100° Fahr. by one pound of combustible. | Pounds of combustible consumed per hour per square foot of grates. | Temperature in degrees Fahr. of the products of combustion in the uptake. | Pounds of water evaporated from 100° Fahr. by one pound of combustible. | Pounds of combustible consumed per hour per square foot of grates. | Temperature in degrees Fahr. of the products of combustion in the uptake. | Pounds of water evaporated from 100° Fahr. by one pound of combustible. | Pounds of combustible consumed per hour per square foot of grates. | Temperature in degrees Fahr. of the products of combustion in the uptake. |
| Table No. 1. Locust Mountain anthracite. Heating surface and calorimeter diminished by suppression of upper horizontal rows of tubes, | 9·070 | 9·657 | 431·5 | 9·696 | 11·024 | 544·4 | 10·592 | 8·116 | 541·3 | 10·814 | 8·602 | 616·1 |
| Table No. 2. Blackheath anthracite. Heating surface and calorimeter diminished by suppression of lower horizontal rows of tubes, . | 10·112 | 11·468 | 460·2 | 9·341 | 9·953 | 453·2 | 10·110 | 10·954 | 577·6 | 10·416 | 10·589 | 603·2 |
| Table No. 2. Blackheath anthracite. Heating surface and calorimeter diminished by suppression of vertical rows of tubes, . . | 10·112 | 11·468 | 460·2 | 9·406 | 10·024 | 477·8 | 9·754 | 11·359 | 567·6 | 10·015 | 10·901 | 597·1 |
| Means, . . | 9·765 | 10·864 | 450·6 | 9·481 | 10·834 | 491·8 | 10·152 | 10·143 | 562·2 | 10·415 | 10·031 | 605·5 |

Referring to the means in the above table, it appears that, with the exception of the experiments with two rows of tubes suppressed, which gave a less economic evaporation of 3 per centum, the economic evaporative efficiency slightly increased with each diminution of tube surface and calorimeter, until, when four rows were suppressed, it was 6¼ per centum greater than with all the tubes in use. The temperature of the products of combustion in the uptake regularly increased with each suppression of tube surface and calorimeter. The rate of combustion remained nearly the same throughout, the extrêmes varying less than 8 per centum.

From these experiments the paradoxical result appears that, in the same boiler with the same furnaces and grate surface, and with the same rate of combustion of the fuel, diminutions of the heating surface, though attended with increased temperatures of the products of combustion in the uptake, effected increased economic evaporative efficiency. At first sight, it seems difficult to reconcile the facts of a less area of the same surface, exposed to the same quantity of heat in the same time, evaporating a greater quantity of water. Nor could this have been the result had the calorimeter been the same in all the cases: it decreased, however, in the same ratio with the tube surface. The calorimeter was the sole variable

element affecting the combustion in the furnaces, and the paradoxical result was a consequence of its variation, and came out in spite of the diminished heating surface. Could the stopped up tubes, in the different experiments, have been added to the tubes in use, as longitudinal extensions of them, the temperature of the products of combustion in the uptake would have been materially reduced, and a corresponding increase of economic evaporative efficiency obtained.

It may be laid down as an indisputable inference that, with equal rate of combustion and equal temperature of the products of combustion in the furnace, increased temperature of these products in the uptake *must* be attended with lessened economic evaporative efficiency. Now, as it was just as indisputably ascertained by experiment that increased temperature of the products of combustion in the uptake *was* attended with increased economic evaporative efficiency, it follows that, in these cases, the temperature of the products of combustion in the furnace must have increased also, and in a still higher ratio. How could this increased temperature of the products of combustion in the furnace have been caused? and how could the cause have been governed by the calorimeter? The answer to these questions is the solution of the paradox.

The temperature of a furnace—that is, of the products of combustion in a furnace—is regulated by the proportion of air admitted to the weight of fuel consumed in a given time, the maximum temperature being obtained when the quantity of air is exactly enough for the maximum oxidation of the fuel. If more air be admitted the temperature falls in nearly a direct proportion. For example: With a purely carbonaceous fuel supplied with just air enough to completely oxidize it, the temperature of the resulting carbonic acid gas will be 4347° Fahr.; if, however, twice this quantity of air be supplied the temperature will be only 2347° Fahr. Nevertheless, the quantity of caloric generated is exactly the same in both cases; but in the last it is diffused through a greater quantity of matter, and consequently proportionably lessened in intensity. In both cases, too, could the products of combustion be cooled down by the absorbing substance to the temperature the fuel had when introduced into the furnace, equal *useful* calorific effects would be obtained. For example: If water were the absorbing substance and supplied at a less temperature than that of the fuel, the products of the combustion of equal weights of the fuel in the two cases would raise the temperature of equal weights of water through 1 degree. But this economic equality is far from being the fact when the temperature of the products of combustion on leaving the absorbing substance, is greater than the temperature of the fuel on its introduction into the furnace. For example: With pure carbon consumed with a supply of air just sufficient to completely oxidize it, the initial temperature of the products of combustion being 4347° Fahr., and the final temperature 500° Fahr., the loss of caloric would be 11·5 per centum, the portion utilized being 88·5 per centum. If, now, twice the above quantity of air be admitted, the initial temperature of the products of combustion being 2347° Fahr., and the final temperature 500° Fahr., as before, the loss of caloric will be 22·3 per centum, the portion utilized being only 77·7 per centum. The useful calorific effects from the same weight of fuel would then, in these two cases, compare as 88·5 to 77·7, and be caused wholly by the mere difference in the quantity of air admitted during the combustion.

Now, in a steam boiler, the proportion of air admitted to the weight of coal consumed on the grates is regulated by the calorimeter; hence the controlling influence of the latter upon the furnace temperature, and, consequently, upon the economic evaporation. In proof of this, let us take, for the sake of simplicity of conception, the case of a boiler in which the temperature of the products of combustion in the uptake remains the same, whether all the tubes be in use or whether one-half of them be stopped up; and let us suppose, further, that in both cases the same weight of the same coal is consumed in the same time; it is evident, then, that with one-half the tubes in use the quantity of air supplied to the coal will be just one-

half of what will be supplied with all the tubes in use, because the quantity of air supplied in a given time depends on the pressure in the furnace and on the aggregate cross area of the tubes or calorimeter. Now, the pressure in the furnace, under the above condition of equal temperature of uptake or chimney, must be the same in both cases; for, *ceteris paribus*, it is the difference between that temperature and the temperature of the external atmosphere which causes the pressure. And with the same pressure and half the area of efflux, it is evident but half the bulk of gases can be discharged or enter; and if but half can enter while the same weight of coal is consumed, the temperature will be nearly double, and the same weight of coal will be consumed, provided half the tubes allowed air enough to enter to completely oxidize it. This effect cannot be produced by diminishing the supply of air, retaining all the tubes in use, either by partially closing ash-pit doors or a damper in the chimney: it results solely from diminution of the calorimeter. Partially closing the ash-pit doors or the chimney damper, controls *the rate of combustion* in the furnace, but not *the quantity of air admitted for the same rate.*

It may be urged that by raising the initial or furnace temperature, the final or uptake temperature will be correspondingly raised, especially in the cases, like the experimental ones, where the increased furnace temperature was attended by a diminished heat-absorbing surface, and so leave the economy of fuel where it was before; but it is easy to show that this cannot be the result, because the efficiency or heat-transmitting power of the same surface increases with the furnace temperature, the temperature of the water in contact with it remaining the same; consequently, not only is the available heat from the fuel increased by diminishing the calorimeter, but less surface is required to absorb it. It is probable, from experiment, that when the difference between the temperature of the products of combustion on one side the heat-transmitting surface of a boiler, and of the water on the other side is doubled, double the quantity of heat will be transmitted in equal times, and so on for other rates of difference. To illustrate the effect of this law upon the efficiency of a given area of heat-transmitting surface, let us suppose the two cases in which the furnace temperature is 4347° Fahr. and 2347°, the temperature of the water in both being 247° Fahr. Then the differences of these temperatures in the two cases is $(4347 - 247 =) 4100°$ and $(2347 - 247 =) 2100°$, which are to each other in the proportion of $\left(\frac{4100}{2100} = \right)$ 1·952 to 1·000; consequently, in the former case, one square foot of surface will, in the same time, transmit nearly twice as much heat as in the latter case, and half the surface will suffice to reduce the temperature of the uptake to the same as in the former case. With a diminished calorimeter, therefore, we have the choice of taking its advantage either in economy of fuel or of heat-transmitting surface. The limit of advantageous diminution of the calorimeter will be reached when it admits too little air for the *perfect* combustion of the fuel, and carbonic oxide instead of carbonic acid gas results from the oxidation of the carbon.

To return to the mean results of all the experiments, we find in them a confirmation of the above theoretical views. With a reduction of the calorimeter of nearly one-half, and of the heat-transmitting surface of over one-third, the rate of combustion was very nearly the same (4·13 per centum less) and the economic evaporation 6¼ per centum greater. The temperature of the products of combustion in the uptake, under these conditions, had risen from 450·6° Fahr. to 605·5°.

# EXPERIMENTS

MADE BY ORDER OF THE NAVY DEPARTMENT

WITH

# THE MACHINE SHOP BOILER

OF THE

## NEW YORK NAVY YARD.

2 P

# EXPERIMENTS

## MADE BY ORDER OF THE NAVY DEPARTMENT

WITH

# THE MACHINE SHOP BOILER OF THE NEW YORK NAVY YARD,

## ON THE ANTHRACITES, SEMI-BITUMINOUS, AND BITUMINOUS COALS FROM THE PRINCIPAL LOCALITIES MINED FOR THE NEW YORK AND PHILADELPHIA MARKETS IN 1862, TO DETERMINE THEIR COMPARATIVE EVAPORATIVE EFFICIENCY.

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THE following experiments were made by a Board composed of Chief Engineers ZELLER, GARVIN, and FITHIAN, convened at the New York Navy Yard by order of the Navy department to ascertain by direct experiment, on a scale of sufficient magnitude, with the requisite accuracy to give precise results, and under the conditions of ordinary practice, the comparative evaporative efficiencies of the anthracites, semi-bituminous, and bituminous coals supplied to the markets of New York and Philadelphia, and generally to the cities of the Atlantic seaboard in 1862, with a view of selection for the use of the Navy.

The members of the Board were instructed to exercise the most rigorous scrutiny in obtaining the samples of coal to be experimented on, and to personally satisfy themselves that each sample was genuine, and of the average quality furnished for general consumption, not a selected lot or a fraudulent substitute from a mine of reputation, and to include in their experiments all the different coals that could be obtained.

The boiler used in making these experiments was the one belonging to the machine shop of the New York Navy Yard. A description and engraving of it have been already given in the immediately preceding article; it is, therefore, necessary to repeat here only that it is of the ordinary horizontal fire-tube type with the tubes returned above its two furnaces, and having a diameter of 3 inches and a length of 8 feet 3 inches.

## COALS.

The coals experimented on comprise anthracites from eleven well-known localities; semi-bituminous coals from two localities, and bituminous coal from one locality. Seven of the anthracites are from Luzerne County, Pennsylvania; the remaining four are from the adjacent County of Schuylkill.

Of the two semi-bituminous coals, one is from Huntingdon County, Pennsylvania; the other is from St. Georges Creek, Alleghany County, Maryland.

The bituminous coal is from Clinton County, Pennsylvania.

The ANTHRACITES were all of essentially the same character and behavior in the furnace. Those from Luzerne County were somewhat harder and possessed more cohesion than those from Schuylkill County; they were a little slower in burning too. The chemical constitution of all was the same, and exclusive of ash and the water mechanically present in the pores, was, by weight, as follows, as appears from the mean of a considerable number of organic analyses made at different times by different persons:

Carbon, 93·89 Hydrogen, 3·55 Oxygen, 2·56

The specific gravity averages 1·55 and the hydroscopic water 2 per centum. It is wholly free from bitumen.

The anthracites kindle slowly. The flame developed is small in quantity and lasts but a brief time, and only at the commencement of the combustion; it is, in color, bluish at first, which soon changes to a faint yellow and then disappears. During combustion the lumps retain their form to the last, consuming away slowly at their surfaces, without softening, swelling, or splitting into fragments; no smoke is emitted at any time under any system of fueling.

In mechanical structure the anthracites are very compact and hard, and possess great cohesion. When struck they give a clear ringing sound, and break with a semi-conchoidal fracture which presents a brilliant jet of almost metallic lustre.

The SEMI-BITUMINOUS coal ignites readily and burns freely with a red flame of moderate length, giving off a moderate quantity of semi-transparent brown smoke. While parting with its volatile elements the lumps lose their form, agglutinate slightly, and intumesce considerably, leaving a bulk of coke much exceeding that of the coal it was derived from. At the commencement of its combustion it cakes on top and requires frequent slicing. Its average specific gravity is 1·42.

The mean of a number of organic analyses gives the following for its chemical composition, exclusive of ash and hygrometric moisture, which latter is about 1½ per centum:

Carbon, 92·37 Hydrogen, 5·07 Oxygen, 2·66

Although the chemical constitution so closely approaches that of the anthracites, yet the mechanical structure is very different, being chiefly columnar, running occasionally into slaty. The columnar portions are semi-crystalline, while the slaty ones are amorphous; the former exhibit a deep shining jet, the latter show a dull black. The cohesion is weak; the slightest blow causes fracture, and the surface is frequently striated; the smallest mechanical action crumbles the lumps into fine powder. The clinker is heavy, and of a dark brown color. It is interspersed with large black, smooth, and shining projections of a vitreous substance. It formed all over the grates and required the fires to be cleaned in the same manner as with the anthracites.

The BITUMINOUS coal ignites easily and quickly, burns with great rapidity, and emits considerable quantities of dense black smoke. It swells about one-third of its original bulk, and works easily in the furnace with but little labor. The clinker is light and porous, and of a light color. It forms in detached masses, has no vitreous character, and in cleaning fires it was only necessary to slice moderately and haul it out with a hook.

This coal has very little cohesion; the slightest blow crumbles it to dust, the fragments being of a cuboidal form. A large lump thrown into the furnace rapidly crumbles to pieces.

Both the semi-bituminous and the bituminous coals used for the experiments were in very small pieces—almost dust.

## MANNER OF MAKING THE EXPERIMENTS.

The boiler was situated on and above the ground floor of the machine shop building. It could be viewed all around, and any leak immediately discovered. It was carefully tested and found to be perfectly tight.

The evaporation was performed under the atmospheric pressure, the steam escaping freely through a pipe of the area of the steam stop-valve, which pipe was bolted to the flange of the stop-valve chamber, on the top of the boiler, and led to the outside of the building on a descent of about 10 degrees from the horizontal, so that any water in it resulting from the condensation of steam by its surface could not return to the boiler. The stop-valve was removed from its chamber and a piece of board bolted to the bonnet flange, leaving the passage for the escape of the steam free from any obstruction whatever.

The feed-pipe for this boiler was in common with two other boilers, and to prevent the possibility of any water entering through it, the flange joint next to the feed check-valve was broken, and a piece of iron bolted tightly over the aperture.

The bottom blow-valve was secured tight in its chamber at the commencement of each experiment, and the pipe attached to it was frequently examined during each experiment, but was never found to have changed its temperature. At the end of each experiment all the water was run out of the boiler to expedite the cooling, so that the tubes, smoke connexions, combustion chambers, &c., could be properly swept of soot and ashes. During all the experiments the water was maintained at 8 inches above the tubes.

To the boiler was attached a glass water gauge of the usual construction, by means of which the height of water could be determined with absolute precision.

The temperature of the products of combustion in the boiler uptake, as they emerged from the tubes was ascertained by a mercurial thermometer which was immersed in a copper pot filled with sand and suspended opposite the centre of the mass of tubes belonging to the furnace nearest to the chimney flue. A small door in the uptake door permitted the pot to be brought near enough to read the thermometer without withdrawing it from the uptake.

The feed-water, previously to being delivered into the boiler, was accurately measured in a tank made of heavy oak plank smoothly lined with sheet lead. The tank was placed on the second floor of the machine shop building, and immediately over the boiler, upon three sleepers of 3 inches depth, the tops of which were leveled; the bottom of the tank, on the inside, was also accurately leveled. A globe-valve of 1½ inch diameter was firmly secured to the tank as near its bottom as practicable, and from it descended the feed-pipe, also of 1½ inch diameter, to a duplicate globe-valve bolted on the front of the boiler just

below the level of the grate-bars. With this valve the attendants in the fire room were enabled to regulate the supply of feed-water so as to maintain it constant at the given mark on the glass gauge. The office of the upper valve was merely to shut off the tank while it was being filled. It will be observed that the entire tank and feed-pipe were in full view, and that the slightest leakage would be instantly visible.

In graduating the tank, both valves in the feed-pipe were opened and the water allowed to descend to its lowest level. A varnished stick was then placed vertically in the tank and deeply marked at the surface of the remaining water: from this zero mark the stick was marked and numbered every inch up to the top of the tank. During an experiment the tank was never refilled until the surface of the water remaining in it was at the zero mark. A straight-edge, accurately leveled, was placed across the tank and secured in position, by screws, at the proper height above the zero mark to include a capacity of precisely 50 feet. A pint measure was used to exactly adjust the water level to the straight-edge each time the tank was filled.

The tank was filled by gravity, through a hose with stop-cock at its end, from the Navy Yard reservoir. As soon as filled the hose was thrown out to avoid the possibility of any water leaking into the tank while it was being discharged.

Every pound of coal thrown into the furnaces was accurately weighed on one of Fairbanks' platform scales, quite new and carefully tested. The tub in which the coal was weighed was of iron; it was precisely counterbalanced on the scales, filled each time to the same weight, and noted the moment it was emptied. In the case of the anthracites, the lumps were made of uniform size of about 3 inches cube, and only lumps were used.

The refuse from the coal in ashes, clinker, soot, and dust was weighed upon the same scales, in the same manner, and in the dry state.

The experiments were conducted in the following manner, namely:—

Before commencing each experiment every part of the fire surface of the boiler was swept clean of soot, dust, and ashes, and the boiler was filled anew with water from the reservoir to the proper height. The water was raised to the boiling point with wood, which was then allowed to burn down to the few embers required to kindle the coal. The height of the water in the boiler was now adjusted on the glass gauge, the tank filled, the time noted, the coal fired, and the experiment held to commence. At the end of the experiment the water was adjusted, by admission, to precisely the same level in the boiler as at the commencement.

Each experiment continued exactly 72 consecutive hours, during which the fires were maintained at the thickness of 7 inches on the grates, kept carefully leveled and free from holes. They were cleaned at regular intervals, and in the same manner and to the same extent. As nearly as possible equal weights of coal were fired per hour, so as to make the rate of combustion not only uniform for the same experiment, but equal for all. The holes in the furnace doors were kept open throughout. For the last hour of the experiment the coal was allowed to burn down, so that but little remained on the grates at its close; and the moment the 72 hours expired the fires were drawn and the unconsumed coal picked out from the ashes, weighed separately, and deducted from the amount expended. The refuse in ashes, clinker, &c., was also weighed, together with the sweepings of the fire surface of the boiler, and the amount added to what was withdrawn from the ash-pits and furnaces during the experiment.

Every care was taken to have all the conditions in the different experiments as nearly alike as possible, and the same experienced firemen and attendants were employed throughout.

TABLE—C OF THE
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| | OUNTAIN. |
| Commercial name of coa | |
| Date of commencing th | t. 14, 1862. |
| State of the weather, | reezes and ar. |
| Duration of the experim | 72 |
| Cubic feet of water evap | 66·489 |
| Pounds of water evapor in the tank, | 73·471 |
| Pounds of coal consume | 82· |
| Pounds of refuse from th | 54· |
| Pounds of combustible c | 78· |
| Per centum of the coal | 13·76 |
| Pounds of coal consume face, . . | 11·085 |
| Pounds of combustible grate surface, . | 9·559 |
| Temperature in degree the shade, . | 59·0 |
| Temperature in degrees | 70·0 |

During each experiment a tabular record was kept, in which was noted hourly the temperature in degrees Fahr. of the external atmosphere in the shade, of the fire room, of the water in the tank, and of the products of combustion in the boiler uptake; also the barometer, and the number of pounds of coal thrown into the furnaces during that hour. In appropriate columns were entered the number of pounds of refuse in ashes, clinker, &c., withdrawn from the ash-pits and furnaces, and the exact time at which each tankful of water was emptied. The condition of the weather was also noted, and the time each fire was cleaned.

The experiments were superintended by four assistant engineers of the Navy, who stood regular watches of six hours each, kept the record, and personally directed all the operations. Each experiment was commenced and ended by Chief Engineer ZELLER, who also supervised it from time to time. It is believed the results are entitled to implicit confidence.

## RESULTS.

In the following table will be found the data and results of the experiments. It is so complete in itself, and so full and clear in its headings and items as to require but little explanation.

It will be observed the experiments are not arranged in the order of the time they were made, but are grouped according to the kind and localities of the coals. The time was governed by conditions growing out of the procuring of the coal.

In the table there are given the total number of cubic feet of water evaporated during each experiment from the temperature of the feed-water in the tank. This quantity is from direct measurement in the tank. From it are derived the corresponding number of pounds of water evaporated from the same temperature, by taking the weight of a cubic foot of water to be 62·321 pounds avoirdupois at the temperature of 62° Fahr., and modifying it, according to the tables of KOPP, for the temperature it had in the tank.

The total number of pounds of coal consumed during each experiment, and the number of pounds of refuse from it in ashes, clinker, dust, and soot, are given and are from direct weighing. The pounds of combustible consumed are the remainder after deducting the refuse from the coal. It includes, of course, the hygrometric moisture of the coal, the oxygen, and doubtless some traces of sulphur and nitrogen, as well as its volatile hydro-carbons and solid carbon.

The rate of the combustion is given both in pounds of coal and in pounds of combustible consumed per hour per square foot of grate surface. As the time and the grate surface were the same for each experiment, these rates are expressed, comparatively, for the different experiments by the total weights of coal and of combustible consumed. It is given, however, per square foot of grate surface per hour for the purpose of comparison with other experiments, and because this is the conventional unit.

In order to compare the economic evaporation given by the different coals, it is necessary that their experimental evaporation be corrected for differences of temperature of feed-water. Accordingly there will be found in the table the total number of pounds of water that would have been evaporated by the total number of pounds of coal and of combustible actually consumed, had the temperature of the feed-water been 100°, and 212° Fahr. instead of the temperature of the tank. It is given for the temperature of 100° because that is about the average temperature of the feed-water with condensing engines, and is convenient for comparison with other experiments. It is also given for the temperature of 212°, as the results from this temperature are the true evaporations, those from lower temperatures being inclusive of

the heating of the water from those temperatures to that of evaporation, as well as of evaporation itself. In deriving the evaporations from the temperatures of 100° and 212° Fahr. of the feed-water, from the experimental temperatures of the feed-water in the tank, the total heat of the steam has been taken from REGNAULT'S tables, and for pressures corresponding to the average height of the barometer during each experiment.

The economic evaporation is given in pounds of water evaporated from the temperatures of 100° and 212° Fahr. by one pound of coal, and by one pound of combustible. The comparative economic evaporative efficiency of the pound of coal, and of the pound of combustible, is also given proportionally, that by the Blackheath anthracite being taken for unity.

In observing the per centum of the anthracites in refuse, it is found to vary from 10·80 to 20·95, a difference of, say, 10 per centum. But while these different per centums are true for the samples actually experimented with, they cannot be, therefore, assumed as true for all the coals from these different mines. The per centum of refuse is so accidental a proportion in coal, and so variable in different samples of the same coal, that a difference of 10 per centum is within the limits of experience with the same coal.

The average per centum of refuse from the anthracites of Luzerne County is 16·25 per centum; and from the anthracites of Schuylkill County 17·63 per centum. The average per centum for all the anthracites is 16·72.

The average per centum of refuse from the semi-bituminous and bituminous coals is 11·82.

The average refuse from anthracite burned on board sea steamers under the conditions of actual practice is from 18 to 20 per centum; from the semi-bituminous and bituminous coals of Pennsylvania and Maryland, east of the Alleghany mountains, it is from 12 to 14 per centum. With, therefore, equal evaporative efficiency per pound of combustible, these latter coals will have a higher economic evaporation, per pound of coal, than the anthracites, and in a higher proportion than (100 — 13 =) 87 to (100 — 19 =) 81, which is the proportion of the remaining combustible.

The per centum of refuse in a coal affects its economic evaporation in a higher degree than the direct ratio of the combustible. For example, in two samples of the same coal, one giving 10 per centum and the other giving 20 per centum of refuse, the economic evaporation will not compare as (100 — 10 =) 90 and (100 — 20 =) 80; the difference will be greater than is expressed by those numbers and for the following reasons. The refuse is chiefly the earthy matter that was mechanically mixed with the coal and has been changed by the action of heat both in its chemical constitution and in its mechanical structure. To produce these changes a certain amount of heat has been absorbed, which amount was obtained from a portion of the combustible, leaving only the remainder for steam generating purposes, supposing the refuse left the furnace with the temperature at which it entered. But as there is a great difference between the temperatures of this earthy matter when it enters the furnace and when it is withdrawn, there is a further loss of heat caused by it and measured by the difference of these temperatures, the specific heat of the substance and its weight.

On comparing, in the different experiments, the temperatures of the products of combustion in the boiler uptake, it will be observed they vary considerably. For the anthracites the minimum was 378·7° Fahr., and the maximum 533·1° Fahr. This great difference could only have been caused by the admission, in the different experiments, of different proportions of air to the weight of coal consumed; and it proves the difficulty of securing, even with the most complete system of precaution, the same conditions throughout a lengthened series of evaporative experiments.

If we compare the temperatures of the products of combustion in the boiler uptake with the evaporative effect, dividing the anthracite experiments into three groups, including in the first all the temperatures below 400° Fahr.; in the second all those between 400 and 425°; and in the third all those above 425°; we shall obtain the following for the mean results, namely:—

| Temperature of Products of Combustion in the Boiler Uptake. | Pounds of Water evaporated from Temperature of 212° Fahr. by One Pound of Combustible. |
|---|---|
| 383·8 | 10·190 |
| 416·0 | 10·791 |
| 451·7 | 10·881 |

From which it appears that the higher the temperature of the *products of combustion in the boiler uptake*, the greater was the economic evaporation. It must not, however, be supposed that these two facts stand to each other in the relation of cause and effect; on the contrary, they are both the effects of a common cause, namely:—the higher temperature of the products of combustion in the furnace produced by the supply of air more nearly approaching the quantity required for the *exact* oxidation of the elements of the coal.

With the semi-bituminous and bituminous coals, the temperature of the products of combustion in the boiler uptake was much higher than with the anthracites; its minimum was 549·5° Fahr., and its maximum 611·1° Fahr. The mean temperature of these products for the anthracites was 425·8° Fahr.; for the semi-bituminous and bituminous coals 581·4° Fahr. The gaseous products of combustion from a given weight of the semi-bituminous and bituminous coals have a greater bulk than those from the anthracites, and require, consequently, a greater calorimeter; more air is also required for the complete oxidation of the elements of those coals. Hence the same calorimeter, with equal weights of coal consumed, will give with the semi-bituminous and bituminous coals a higher furnace temperature and, likewise, a higher uptake temperature.

On comparing the economic evaporative efficiency of the anthracites, it appears that the minimum was 11·3 per centum below the maximum, taking the latter for unity. This difference is within the limits of errors of observation and, principally, of variations in the conditions of the experiments, which can neither be avoided nor even discovered in making them, and which only appears in the results, as we have just seen examples of in the different temperatures of the products of combustion in the uptake. Had each experiment been made upon, say, fourteen hundred tons of each kind of coal, instead of about fourteen tons, and mined at different times, and from different portions of the same vein, it seems probable that the economic evaporative efficiency would have been sensibly equal. In the matter of the evaporative power of a coal from any locality, a sound deduction can only be drawn from the mean of the combustion of a very large quantity, taken from different parts of the mine so as to be a fair representative of the mean *quality* of the coal, and experimented with at different times so as to obtain a fair average of *conditions*. Let us apply this rule as far as possible to the samples of anthracite experimented on, comparing the results given by those from Luzerne County with the results given by those from Schuylkill County, namely:—

| | Per centum of the coal in ashes, clinker, soot, and dust. | Temperature of the Products of Combustion in the Uptake. | Pounds of water evaporated from temperature of 100° Fahr., by one pound of Combustible. |
|---|---|---|---|
| Anthracites from Luzerne Co. (8 samples) | 16·25 | 446·1° Fahr. | 9·601 |
| Anthracites from Schuylkill Co. (4 samples) | 17·63 | 410·2° " | 9·532 |

These results may be considered practically equal, and could the number of samples have been doubled, the slight differences now observed would have been greatly lessened or have altogether disappeared. The experiments, therefore, entitle us to conclude that, in general, the economic evaporative efficiency of the anthracites from the different localities experimented on, are equal; though particular samples may give results whose extreme variation is as high as, say, 10 per centum, or 5 per centum on each side of the mean.

From the same considerations it will appear that the economic evaporative efficiency of the semi-bituminous and bituminous coals is also equal, the extreme difference being less than 8 per centum of the highest. The means are as follows, namely:—

| | Per centum of the coal in ashes, clinker, and soot. | Temperature of the Products of Combustion in the Uptake. | Pounds of water evaporated from temperature of 100° Fahr., by one pound of Combustible. |
|---|---|---|---|
| Semi-bituminous, (2 samples,) | 11·13 | 580·3° Fahr., | 10·230 |
| Bituminous, (1 sample,) | 13·03 | 583·7° " | 9·589 |

Comparing these figures with the corresponding ones from the anthracites, it appears that the economic evaporation by the pound of combustible from the bituminous and anthracite coals is equal; while that by the pound of combustible from the semi-bituminous and anthracite coals shows a difference of nearly 6½ per centum in favor of the former, taking its result for unity.

The semi-bituminous and bituminous coals had a more favorable condition than the anthracites in the higher furnace temperature—evidenced by the higher temperature of the products of combustion in the boiler uptake—due to the fact that the calorimeter of the boiler was better proportioned for them, owing to the greater bulk of their gaseous products of combustion, and greater quantity of air required. With equal furnace temperature and rate of combustion, the economic evaporation by the pound of combustible of the semi-bituminous coals and of the anthracites will be equal; and that by the latter will exceed the economic evaporation by the pound of combustible from the bituminous coal by, probably, 6 or 8 per centum.

## EXPERIMENT MADE ON THE GLEN CARBON SEMI-ANTHRACITE, FURNISHED FROM THE MINE OF B. HAMMITT, SITUATED IN THE WESTERN PORTION OF THE PENNSYLVANIA COAL FIELD, TO DETERMINE ITS ECONOMIC EVAPORATIVE EFFICIENCY WITH THE BOILER OF THE MACHINE SHOP OF THE NEW YORK NAVY YARD.

After the termination of the previously recorded experiments with the Machine Shop Boiler of the New York Navy Yard on different coals; another was made on the Glen Carbon semi-anthracite from the mine of B. Hammitt.

The Glen Carbon semi-anthracite is from the western portion of the Pennsylvania Middle Coal Field. In the furnace it burns freely with a long yellow flame; makes but little clinker, and that of a scoriacious character, not adhering to the grate bars but leaving them in small particles. It does not intumesce nor agglutinate; and it breaks up easily into small pieces. Its cohesion is less than that of the true anthracites; and it soils the hand more.

The sample experimented with was furnished by the contractor for the regular coal supplies, and was burned in the condition delivered.

In the following table will be found the data and results of the experiment, which was conducted in all respects in precisely the manner already described for the preceding experiments.

TABLE CONTAINING THE DATA AND RESULTS OF THE EXPERIMENT MADE ON THE GLEN CARBON SEMI-ANTHRACITE TO DETERMINE ITS ECONOMIC EVAPORATIVE EFFICIENCY WITH THE MACHINE SHOP BOILER OF THE NEW YORK NAVY YARD.

| | | |
|---|---|---|
| | Date of commencing the experiment, | 10.15 A.M. Dec. 2, 1862. |
| | State of the weather, | Clear with light breeze. |
| TOTAL QUANTITIES. | Duration of the experiment in consecutive hours, | 60· |
| | Number of cubic feet of water evaporated, | 3,810·8 |
| | Number of pounds of water evaporated, | 237,736·059 |
| | Number of pounds of semi-anthracite consumed, | 30,242· |
| | Number of pounds of refuse from the semi-anthracite in ashes, &c., | 5,342· |
| | Number of pounds of combustible consumed, | 24,900· |
| | Per centum of the semi-anthracite in refuse, | 17·66 |
| MEAN QUANTITIES. | Temperature in degrees Fahr. of the external atmosphere in the shade, | 41·2 |
| | Temperature in degrees Fahr. of the fire room, | 61·2 |
| | Temperature in degrees Fahr. of the feed-water in the tank, | 46·4 |
| | Temperature in degrees Fahr. of the products of combustion in the boiler uptake, | 542·1 |
| | Barometer, | 30·18 |
| | Number of pounds of semi-anthracite consumed per hour per square foot of grate surface, | 14·001 |
| | Number of pounds of combustible consumed per hour per square foot of grate surface, | 11·528 |
| EVAPORATION. | Number of pounds of water evaporated from a temperature of 100° Fahr., | 249,554·509 |
| | Number of pounds of water evaporated from a temperature of 212° Fahr., | 278,482·376 |
| | Pounds of water evaporated from a temperature of 100° Fahr. by one pound of semi-anthracite, | 8·252 |
| | Pounds of water evaporated from a temperature of 100° Fahr. by one pound of combustible, | 10·022 |
| | Pounds of water evaporated from a temperature of 212° Fahr. by one pound of semi-anthracite, | 9·208 |
| | Pounds of water evaporated from a temperature of 212° Fahr. by one pound of combustible, | 11·184 |

# EXPERIMENTS

MADE WITH THE

# MACHINERY OF THE U. S. IRON-CLAD STEAM BATTERY

# "MONITOR."

# EXPERIMENTS

MADE WITH THE

# MACHINERY OF THE U. S. IRON-CLAD STEAM BATTERY "MONITOR,"

TO DETERMINE THE COST OF THE INDICATED HORSE POWER IN POUNDS OF STEAM AND OF FUEL PER HOUR WHEN USING THE STEAM EXPANSIVELY AND WITHOUT EXPANSION; AND TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER WITH ANTHRACITE.

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THE United States iron-clad steam battery "MONITOR" having been brought to the Washington, D. C. Navy Yard for repairs to her machinery, and for alterations and additions; advantage was taken of the opportunity thus afforded, after the machinery was put in thorough order, to make with it a set of experiments to determine the cost of the indicated horse power in pounds of steam and of fuel per hour, with the steam used expansively and without expansion. And, also, to determine the evaporative efficiency of the boiler with anthracite.

These determinations were the more valuable owing to the peculiar type of the engine, which involved conditions affecting the economic result in a remarkable manner, and which do not exist with engines of the usual types. The proportions of the boiler were likewise very unusual, and the results are instructive in confirmation of the great influence exerted on the evaporative efficiency by the proportion of calorimeter employed.

The boilers when the vessel arrived at the Navy Yard, had no chimney except two pipes of about 4 feet length above the deck, and depended entirely on the fan blowers driven by small steam cylinders for draught. Of course these blowers could not be started until steam was raised, which was a very slow operation with boilers of remarkably small calorimeter and scarcely any chimney. To remedy this inconvenience a permanent telescopic chimney was added; and by its aid the experiments were enabled to be made; for the natural draught was now sufficient to supply steam enough for that purpose. To have made the experiments with the blowers would have been useless, as there was no means to ascertain what portion of the steam was consumed in driving them; and for the same reason the power of the engines was limited by the quantity of steam that could be furnished by the natural draught,

Before proceeding with a description of the manner of making the experiments, it is necessary, for a clear comprehension of them, to preface with a description of the machinery with which they were made. It will be found as follows, namely:—

## ENGINES.

The engines of the "MONITOR" were designed by JOHN ERICSSON according to his patent, and are styled "vibrating lever engines." The two cylinders are horizontal, both in one casting, and have one bottom in common which is bolted in at its centre. The cylinders lie athwartship and are wholly above the screw shaft. They are bolted to a strong foundation plate which likewise extends athwartship and above the screw shaft. This foundation plate, at each outboard side, is formed into the pillow-blocks of the vibrating lever shafts. The after side of it, at the centre, is also formed into the pillow-block for the end journal of the screw shaft.

The valve-chests are upon the forward side of the cylinders and contain a balanced slide, and an independent slide cut-off valve, each worked by its own eccentric keyed upon a small shaft revolved by the screw shaft, and lying in the prolongation of its axis. The motion is reversed by turning the small shaft with its eccentrics by hand.

The piston has a half trunk, that is a trunk upon its outboard end in place of a piston rod, and is fitted with a pin at the centre. From the pin at the centre of the piston, a connecting link extends within the half trunk to an arm forged on the vibrating lever shaft.

The vibrating lever shaft, one to each cylinder, extends in the fore and aft direction of the vessel, and is supported in two pillow-blocks on the foundation plate. An arm opposite the centre of the cylinder and forged on the shaft receives, by means of the connecting link, the movement of the piston. Another arm, but of greater length, keyed upon the shaft beyond the after side of the cylinders, transmits by means of an ordinary connecting rod, the movement to the crank pin.

There is but one crank and crank pin for both engines. The crank is keyed on the forward end of the screw shaft, and revolves within an opening in the bed-plate, which latter is extended sufficiently abaft the cylinders for this purpose, and to carry the pillow-block of the end journal of the shaft.

There is but one condenser and air-pump for both engines; the condenser is of the common jet kind; the air-pump is horizontal and double-acting, and is worked from an arm on one of the vibrating lever shafts prolonged. The casting containing the condenser, air-pump, channel-ways, hot-well, &c., is bolted independently of the engine to the cross keelsons of the vessel, and is situated abaft the foundation plate, so that the crank revolves between it and the cylinders.

The feed and bilge-pumps are horizontal and single-acting; they are bolted one on each side of the casting containing the condenser and air-pump, and are worked from the air-pump cross-head.

The following are the principal dimensions of the most important parts, namely:—

| | |
|---|---|
| Diameter of the cylinder, . . . . . . | 40 inches. |
| Diameter of the half trunk, . . . . . . . | 13½ " |
| Average area of the piston for both strokes of one cylinder, exclusive of half trunk, . . . . . . . | 1185·07 square inches. |
| Stroke of the piston, . . . . . . . | 22 inches. |
| Space displacement of both pistons per stroke, average of both strokes and exclusive of half trunk, . . . . . . | 30·175 cubic feet. |
| Clearance of the piston, . . . . . . . | ¾-inch. |

| | |
|---|---|
| Space comprised between the main slide-valve and the piston when at the end of its stroke, being contents of steam passage and clearance, | 1·395 cubic foot. |
| Space comprised between the steam and cut-off valves, | 0·835 " |
| Area of steam-port (4 by 16 inches=), | 64 square inches. |
| Area of exhaust (5 by 16 inches =), | 80 " |
| Diameter of air-pump, | 16 inches. |
| Diameter of air-pump piston-rod, | 2¾ " |
| Stroke of air-pump piston, | 18 " |
| Capacity of air-pump, exclusive of piston-rod, | 2·063 cubic feet. |
| Net area of opening at each end of air-pump, for receiving and delivery valves, | 62·5 square inches. |
| Diameter of feed and of bilge-pumps, | 5 inches. |
| Stroke of plunger of feed and bilge-pumps, | 18 " |
| Diameter of journals of vibrating lever shaft, | 6 " |
| Length of journals of vibrating lever shaft, | 9½ " |
| Diameter of vibrating lever shaft at its centre, | 11 " |
| Diameter of main journal of screw shaft, | 9 " |
| Diameter of journal in vibrating lever for main connecting rod, | 4½ " |
| Length of journal in vibrating lever for main connecting rod, | 5 " |
| Diameter of crank-pin, | 5½ " |
| Length of crank-pin journal for each connecting rod, | 5 " |
| Diameter of main connecting rod in neck at crank-pin end, | 3¾ " |
| Diameter of main connecting rod in neck at vibrating lever shaft end, | 3½ " |
| Length of main connecting rod between centres, | 53½ " |
| Diameter of journal of pin in the steam piston, | 4½ " |
| Length of journal of pin in the steam piston, | 7 " |
| Diameter of journal of pin in short arm of vibrating lever shaft, for main link, | 5½ " |
| Length of journal of pin in short arm of vibrating lever shaft, for main link, | 7 " |
| Length between centres of connecting link in the half trunk, | 46½ " |
| Depth of piston at edge, | 7 " |
| Depth of piston at centre, | 10½ " |
| Depth of cast iron packing rings, | 3½ " |
| Length, between centres, of the vibrating lever worked by the steam piston, | 24 " |
| Length, between centres, of the vibrating lever which works the main connecting rod, | 44 " |
| Length of crank, between centres, | 20½ " |

## SCREW.

One true screw of cast iron. The forward edge of the blades are at right angles to the axis of the screw: the after edge tapers in a straight line from the periphery to the hub.

| | |
|---|---|
| Diameter of the screw, | 9 feet. |
| Diameter of the hub, | 1 foot 6 inches. |
| Pitch, | 16 feet. |
| Number of blades, | 4. |
| Length of the screw at the periphery, in the direction of its axis, | 2 feet. |
| Length of the screw at the hub, in the direction of its axis, | 1 foot 8 inches. |
| Thickness of blades above fillet of hub, | 4½ " |
| Thickness of blades at periphery of screw, | 1 inch. |

## BOILERS.

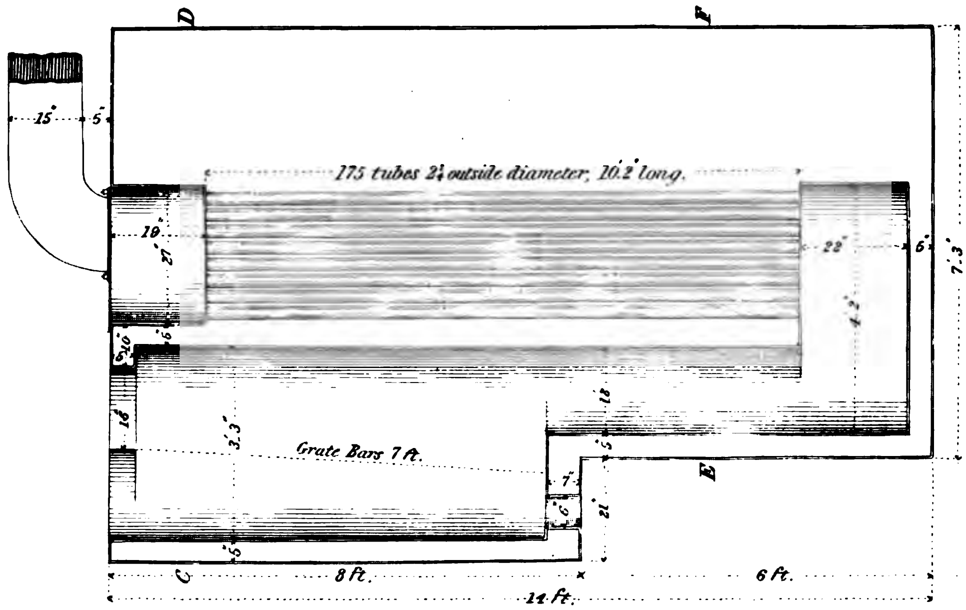
The boilers are of the horizontal fire tube type with the tubes returned above the furnaces. They are two in number, and are placed side by side in the vessel.

The shell is rectangular with the flat top and bottom joined to the vertical sides by quadrantal arcs of 28 inches radius for the top, and 11 inches radius for the bottom. The height of the shell for a length of 8 feet from the front is 9 feet, thence, the bottom being raised by a vertical offset 21 inches, it is 7 feet 3 inches.

Each boiler contains two furnaces. The crowns are flat and joined to the vertical sides by quadrantal arcs of 6 inches radius. The bottoms of the ash-pits are also joined to the vertical sides by quadrantal arcs of 6 inches radius. In the back of each ash-pit is an aperture of 24 inches length and 6 inches width, with semicircular ends, intended for—but never used with—a blast pipe. The height from the bottom of the ash-pit to the crown of the furnace is 39 inches. The top of the grate-bars at the front of the furnace is 18 inches above the bottom of the ash-pit, and at the back of the furnace 13 inches. The opening for the furnace door is 18 inches wide and 16 inches high, with semicircular top. The door is of cast iron and fitted with a register of four holes, exposing an aggregate opening of 7½ square inches for the admission of air into the furnace above the fuel. The lining plate is pierced with the same number of holes, of the same form, and placed immediately opposite.

From the furnace a combustion chamber of 4 feet 4 inches length extends to the back smoke connexion. This chamber is in cross section a parallelogram of 40½ inches width and 18 inches height, with the corners rounded on quadrantal arcs of 6 inches radius.

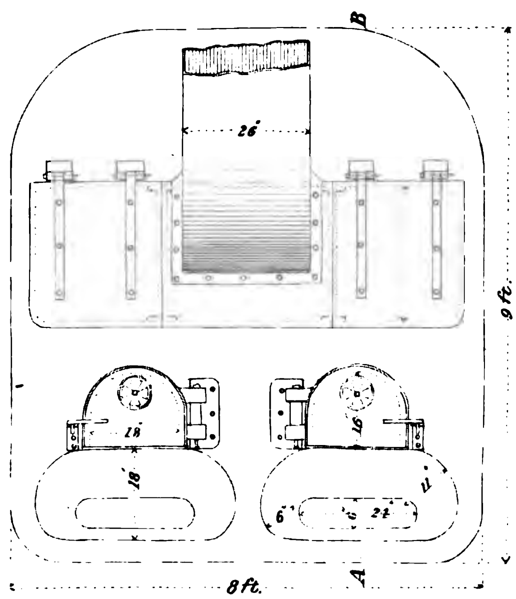
The back smoke connexion—separate for each furnace—is 40½ inches wide, 22 inches deep, and 4 feet 2 inches high. It is flat on the top and bottom which are joined to the sides by quadrantal arcs of 6 inches radius.

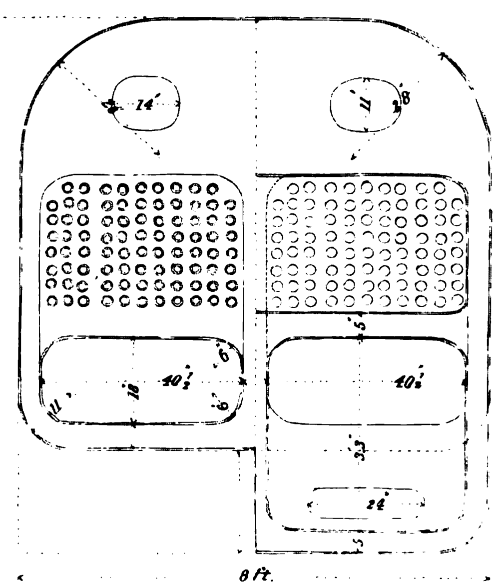



*Longitudinal Section through AB.*

**Boiler of U.S. Iron Clad Steam Battery**

# MONITOR.




*Front Elevation.* *Half Section on EF. Half Section on CD.*

From the back smoke connexion to the uptake, there extend, for each furnace, eighty-six horizontal fire tubes of $2\frac{1}{4}$ inches outside diameter and $\frac{1}{10}$th of an inch thickness of metal. They are in eleven rows horizontally, and eight rows vertically; the two upper corners tubes are omitted. The least space horizontally between the tubes in the clear is $1\frac{1}{4}$ inch, and vertically 1 inch. They are separated in two places horizontally to leave room for vertical stays, and here they are $2\frac{1}{4}$ inches apart in the clear. The total height occupied by the tubes is 25 inches. The distance between the crown of the furnace and the lower tube is 6 inches in the clear.

The uptake, in common for both furnaces, is rectangular in form with the top and bottom joined to the sides by quadrantal arcs of 6 inches radius. Height of uptake 27 inches, depth 19 inches, width 7 feet 2 inches. There are three uptake doors, of which the centre one is fixed, and the remaining two hinged and movable. Upon the centre door is bolted the lower part of the chimney which is rectangular in section and 26 inches by 15 inches. This part rises vertically, enlarges at the deck of the vessel and abuts up to it, the opening in the deck being protected by a grating of wrought iron of sufficient strength to resist shot, and exposing an aggregate area of opening of 390 square inches. Originally, a square pipe 4 feet high above the deck was placed upon the grating and completed the chimney; the blowing engines being depended on for the draught. This, however, being found very inconvenient, and requiring a great length of time to raise steam, the square pipes were removed just previous to the experiments at the Washington Navy Yard, and the two gratings (one for each boiler) connected above deck by a breeching from the centre of which rose a telescopic pipe of $31\frac{1}{2}$ inches least diameter.

All the flat water spaces are 5 inches wide, including both thicknesses of metal. There are two manholes in the steam room, at the back of the boiler, and the bottom of each water leg has a hand-hole.

There is no steam drum, or steam chimney to the boiler, and, with the water at 9 inches above the tubes, there is just two feet between the top of the boiler and the water level. The steam stop-valve is bolted to the front of the boiler, a dry pipe on the inside of the boiler connects with it, and the steam-pipe to the engines leads from it.

The thickness of the plate for the bottom of the boiler at the front for a length of 8 feet, and for the bottom of the ash-pits, is $\frac{7}{16}$-inch. The tube plates are $\frac{1}{2}$-inch thick. All other parts, except the tubes which are $\frac{1}{10}$-inch thick, are $\frac{3}{8}$-inch thick. The boiler is braced for a hydrostatic test pressure of 40 pounds per square inch above the atmosphere.

The following are the principal dimensions and proportions, namely:—

| | |
|---|---|
| Length of each boiler, | 14 feet. |
| Breadth of each boiler, | 8 " |
| Height of each boiler, | 9 " |
| Number of furnaces in both boilers, | 4. |
| Breadth of each furnace, | 3 feet $4\frac{1}{2}$ inches. |
| Length of each furnace, | 7 " |
| Area of total grate surface, | 94·5 square feet. |
| Total number of tubes in both boilers, | 344. |
| Extreme length of each tube, | 10 feet 2 inches. |
| Exterior diameter of each tube, | 2·25 inches. |
| Interior diameter of each tube, | 2·05 " |

| | |
|---|---|
| Total area of heating surface in the furnaces, . . . | 213·36 square feet. |
| Total area of heating surface in the combustion chambers, . . | 154·12 " |
| Total area of heating surface in the back smoke connexion, . . | 187·80 " |
| Total area of heating surface in the tubes, calculated for their inside circumference, . . . . . . . . | 1,876·98 " |
| Total area of heating surface in the uptakes, . . . . | 73·85 " |
| Total area of heating surface in the two boilers, . . | 2,506·11 " |
| Aggregate cross area of the combustion chambers for draught, . | 19·392 " |
| Aggregate cross area of the tubes for draught (inner diameter), . . | 7·887 " |
| Cross area of the chimney, . . . . . . | 5·412 " |
| Diameter of the chimney, . . . . . . . | $31\frac{1}{2}$ inches. |
| Height of the chimney above the grates during the experiment of using the steam expansively, . . . . . . | 42 feet 6 inches. |
| Height of the chimney above the grates during the experiment of using the steam without expansion, . . . . . . | 33 " |
| Distance traversed by the products of combustion from the centre of the furnace to their delivery into the uptake, . . . . . | 22 " |
| Capacity of the steam room above 9 inches above the tubes, . . | 382· cubic feet. |
| Weight of water up to 9 inches above the tubes, . . | 74,000· pounds. |
| Ratio of the heating to the grate surface, . . . . | 26·520 to 1·000. |
| Ratio of the grate surface to the cross area of the combustion chambers, | 4·873 " |
| Ratio of the grate surface to the cross area of the tubes, . . | 11·982 " |
| Ratio of the grate surface to the cross area of the chimney, . . | 17·461 " |

## MANNER OF MAKING THE EXPERIMENTS.

During the making of the experiments the vessel was secured to the wharf of the Washington Navy Yard, D. C.; and the power developed by the engines was employed in forcing the water astern by the screw propeller.

The power developed by the engines was ascertained by two indicators, one permanently attached to each cylinder and not removed during the experiments; and by a self-registering counter worked by the engines. The indicators were placed near the centre of a pipe connecting the two ends of the cylinder; and, by means of suitable cocks, could be placed in communication with either end. They were excellent instruments, graduated to 10 pounds pressure per square inch per inch, and gave true and smooth diagrams.

The cost of the power was ascertained in pounds of anthracite, pounds of combustible, and pounds of feed-water or steam consumed per hour.

The anthracite consumed, and the refuse from it in ashes, clinker, dust, &c., were accurately weighed on a new Fairbanks' platform scales; the refuse in a dry state.

The feed water, before being pumped into the boiler, was measured with scrupulous care in a wooden tank lined with sheet lead, which discharged at each delivery 47·55 cubic feet of water into the boiler.

The tank was placed upon the wharf with its bottom, at all states of the tide, above the deck of the vessel. It was filled from the Navy Yard Reservoir by gravity through a hose with a pipe and stop-cock at the end. As soon as the tank was filled this hose was thrown out to insure against possible leakage into the tank while it was discharging into the boiler. From the bottom of the tank to the receiving nozzle of the engines feed-pump there was connected a leather hose with a stop-cock close to the bottom of the tank. The suction pipe between the feed-pump and hot-well had been removed to permit this arrangement, and the aperture in the hot-well closed by a blank flange. The tank was mounted on supports which allowed it to be viewed all around and beneath. The hose connecting it with the feed-pump was in full view for its whole length; and the feed-pipe from the pump to the boiler, being overhead, was also in full view for its whole length. There was no possibility of the leakage of a drop of water without discovery. The tightness of the boiler was tested by allowing the water to stand in it under the average steam pressure for a number of hours, and no appreciable fall could be observed. It is believed that the entire apparatus of tank, pump, hose, pipes, and boilers were perfectly tight, and that there was no leakage of water from any cause out of them.

The blow-off valves of the boilers had just been ground in tight,—as had indeed all the cocks and valves of the machinery, when the vessel was out of water on the ways,—they were carefully secured closed, and were not opened during the experiments.

The boilers were provided with glass water gauges of the usual construction. The temperature of the injection-water was obtained from a thermometer immersed overboard; and of the feed-water from a thermometer kept constantly immersed in the tank. The temperature of the hot-well was ascertained by drawing water from it upon a thermometer immersed in a pot. The steam pressure in the boilers, and the vacuum in the condenser, were denoted by open mercurial syphon gauges of the usual construction. The height of the barometer was taken from an instrument of the usual construction for the Navy in a neighboring vessel.

The experiments were conducted by the engineer department of the vessel. The firemen were experienced, and all were familiar with the machinery. The assistant engineers were in four watches of four hours each, and personally attended to the tank, the weighing, the taking of the indicator diagrams, and the keeping of the record, &c. The firemen of each watch were regularly posted; each had his special service and nothing more to perform.

Just previous to the experiments the machinery had been entirely overhauled, new brasses put in, new joints made, all cocks and valves ground in, the cylinders' steam and cut-off valves re-faced and scraped to their seats, and the packing of the pistons carefully set up and re-adjusted. It is believed that both valves and pistons were perfectly tight, and that there was no leakage of steam past them.

The experiments were two in number: the first, of seventy-two consecutive hours duration, was made using the steam expansively, cutting it off in the cylinder at 0·425 of the stroke of the piston from the commencement by the fixed cut-off slide. The second, of thirty-three consecutive hours duration, was made using the steam absolutely without expansion, for the steam slide-valve had neither lap nor lead. During this experiment the cut-off valve was working as usual, and steam was given during the whole stroke of the piston by keeping the pass-over valve open. The second experiment would have been continued seventy-two hours, the same as the first, had not the services of the vessel been unexpectedly required.

There was one screw throttle-valve in the main steam-pipe for both engines, and during both experiments it was kept at the same opening, being nearly closed. The boiler pressure was nearly the same throughout, and care was taken in each experiment that the conditions should be as uniform as possible from com-

mencement to end. The steam pressure, vacuum, temperatures, speed of piston, rate of combustion of the fuel, &c., varied between only very trifling limits. The water in the boiler was carried at 6 inches above the tubes, and was fed in very uniformly by means of the cock in the suction hose at the bottom of the tank. The fires were carried 7 inches thick upon the grates, and were kept well cleaned, leveled, and free from holes. The anthracite was burnt with the natural draught, and, although the fires were not forced, yet as much was burned as they would take: the rate of combustion was, therefore, the maximum under the conditions. During the first experiments the chimney was 42 feet 6 inches high above the level of the grates: it had been made this height by an additional piece temporarily applied, which, on account of an accident to it in the interval between the experiments, having to be left off in the second one, reduced the height to 33 feet.

Before commencing each experiment, the machinery was operated about 6 hours to adjust it to the normal conditions required to be uniformly maintained during its continuance, and to bring the fires to steady action. No note was taken of the water and coal thus expended. When all was ready, average fires and steam pressure being in the boilers, and the water in the glass gauge being at the 6 inches above the tubes, the tank was filled, the time and the number on the counter noted, and the experiment commenced. From this time to its end all the quantities were systematically weighed or measured and noted. At the end of the experiment, the water in the boilers was brought to exactly the same level as at the beginning, and the fires, having been thoroughly cleaned, were made the same thickness as at the beginning.

In conducting the experiments a tabular record was kept, in which was entered at the end of each hour in appropriate columns the number on the counter, the average number of double strokes of engines' pistons made per minute during that hour, the average steam pressure in the boilers, the average vacuum in the condenser, the temperature of the injection-water, of the hot-well, of the feed-water, of the engine room, and of the external atmosphere in the shade, and the number of pounds of anthracite thrown into the furnaces during that hour. The weight of refuse and the height of the barometer were noted in their respective columns at the close of each watch of four hours. In another column was recorded the exact time at which each tankful of water was emptied. At the end of every hour an indicator diagram was taken from each end of each cylinder.

The data and results of the experiments made with the machinery, in the manner and under the conditions hereinbefore described, will be found in the following table, namely:—

TABLE CONTAINING THE DATA AND RESULTS OF THE EXPERIMENTS MADE WITH THE MACHINERY OF THE U.S. IRON-CLAD STEAM BATTERY "MONITOR" AT THE WHARF OF THE WASHINGTON NAVY YARD, D. C., TO DETERMINE THE COST OF THE INDICATED HORSE POWER IN POUNDS OF STEAM AND OF COAL PER HOUR, AND THE EVAPORATIVE EFFICIENCY OF THE BOILER WITH ANTHRACITE.

| | | USING THE STEAM EXPANSIVELY. | USING THE STEAM WITHOUT EXPANSION. |
|---|---|---|---|
| | Date of commencing the experiment, | 2 P. M. Nov. 1, 1862. | Midnight. Nov. 7, 1862. |
| | State of the weather, | Clear and dry with light breeze. | Cloudy with mod. breezes, 12 hours snowing. |
| TOTAL QUANTITIES. | Duration of the experiment, in consecutive hours, | 72· | 33· |
| | Number of double strokes made by the engines' pistons, per counter, | 165,326· | 65,479· |
| | Pounds of feed-water pumped into boilers, per tank, | 340,676·418 | 133,667·600 |
| | Pounds of anthracite consumed, | 40,950· | 16,287· |
| | Pounds of refuse from the anthracite, in ashes, clinker, soot, and dust, | 10,200· | 3,817· |
| | Pounds of combustible consumed, | 30,750· | 12,470· |
| | Per centum of refuse from the anthracite, | 24·91 | 23·44 |
| QUANTITIES PER HOUR. | Pounds of feed-water pumped into boilers, | 4,731·617 | 4,050·533 |
| | Pounds of anthracite consumed, | 568·750 | 493·545 |
| | Pounds of combustible consumed, | 427·083 | 377·879 |
| | Pounds of anthracite consumed per square foot of grate, | 6·019 | 5·223 |
| | Pounds of combustible consumed per square foot of grate, | 4·519 | 3·999 |
| ENGINES. | Number of double strokes made per minute by the engines' pistons, | 38·27 | 33·07 |
| | Vacuum in condenser, in inches of mercury per gauge, | 27·4 | 27·0 |
| | Height of barometer, in inches of mercury, | 29·75 | 29·55 |
| | Fraction of the stroke of piston completed when the steam was cut off, | 0·425 | 1·000 |
| | Steam pressure in boilers, in pounds per square inch above atmosphere, per gauge, | 16·6 | 18· |
| | Area of the throttle-valve open, in square inches, | 4· | 4· |
| STEAM PRESSURE IN CYLINDERS, PER INDICATOR. | In pounds per square inch above zero at beginning of stroke of piston, | 16·6 | 11·1 |
| | In pounds per square inch above zero at point of cutting off the steam, | 14·7 | |
| | In pounds per square inch above zero at end of stroke of piston, | 7·6 | 9·2 |
| | In pounds per square inch above zero against the piston during its stroke, | 4·3 | 3·9 |
| | Mean gross effective pressure in pounds per square inch of piston during its stroke, | 8·2 | 6·5 |
| | Mean total pressure in pounds per square inch of piston during its stroke, | 12·5 | 10·4 |
| | Mean net pressure in pounds per square inch of piston during its stroke, | 6·7 | 5·0 |

(TABLE CONTINUED.)

| | | | USING THE STEAM EXPANSIVELY. | USING THE STEAM WITHOUT EXPANSION. |
|---|---|---|---|---|
| POWER. | ABSOLUTE. | Gross effective horses power developed by the engine, | 82·642 | 56·608 |
| | | Total horses power developed by the engine, | 125·980 | 90·573 |
| | | Net horses power developed by the engine, | 67·525 | 43·545 |
| | ECONOMIC. | Pounds of feed-water consumed per hour per gross effective horse power, | 57·254 | 71·554 |
| | | Pounds of feed-water consumed per hour per total horse power, | 37·558 | 44·721 |
| | | Pounds of feed-water consumed per hour per net horse power, | 70·072 | 93·019 |
| | | Pounds of anthracite consumed per hour per gross effective horse power, | 6·882 | 8·718 |
| | | Pounds of anthracite consumed per hour per total horse power, | 4·515 | 5·449 |
| | | Pounds of anthracite consumed per hour per net horse power, | 8·423 | 11·334 |
| | | Pounds of combustible consumed per hour per gross effective horse power, | 5·168 | 6·675 |
| | | Pounds of combustible consumed per hour per total horse power, | 3·390 | 4·172 |
| | | Pounds of combustible consumed per hour per net horse power, | 6·325 | 8·678 |
| TEMPERATURE | IN DEGREES FAHR. | Of the injection-water, | 54·2 | 45·9 |
| | | Of the hot-well, | 105·0 | 87·3 |
| | | Of the feed-water, | 58·8 | 54·3 |
| | | Of the engine room, | 100·8 | 92·0 |
| | | Of the external atmosphere in the shade, | 54·8 | 33·0 |
| TOTAL EVAPORATION. | FROM TEMP. OF FEED-WATER | Pounds of steam discharged from cylinders into condenser, calculated from the pressure of the steam at the end of the stroke of the piston, | 220,732·678 | 104,846·597 |
| | | Pounds of steam equivalent to the heat annihilated in the cylinder to produce the total power of the engine, calculated from JOULE's equivalent, | 23,547·851 | 7,807·916 |
| | | Pounds of steam evaporated from water at 100° Fahr., per INDICATOR, | 253,509·651 | 117,372·540 |
| | | Pounds of steam evaporated from water at 100° Fahr., per TANK, | 353,547·452 | 139,265·666 |
| | | Pounds of steam evaporated from water at 212° Fahr., per INDICATOR, | 282,526·596 | 130,797·504 |
| | | Pounds of steam evaporated from water at 212° Fahr., per TANK, | 394,014·815 | 155,194·745 |

(TABLE CONTINUED).

| | | USING THE STEAM EXPANSIVELY. | USING THE STEAM WITHOUT EXPANSION. |
|---|---|---|---|
| ECONOMIC EVAPORATION. | Pounds of steam evaporated from water at 100° Fahr., by one pound of anthracite, per INDICATOR, . | 6·191 | 7·207 |
| | Pounds of steam evaporated from water at 100° Fahr., by one pound of anthracite, per TANK, . . | 8·633 | 8·551 |
| | Pounds of steam evaporated from water at 212° Fahr., by one pound of anthracite, per INDICATOR, . | 6·900 | 8·031 |
| | Pounds of steam evaporated from water at 212° Fahr., by one pound of anthracite, per TANK, . . | 9·622 | 9·529 |
| | Pounds of steam evaporated from water at 100° Fahr., by one pound of combustible, per INDICATOR, . | 8·244 | 9·412 |
| | Pounds of steam evaporated from water at 100° Fahr., by one pound of combustible, per TANK, . . | 11·497 | 11·168 |
| | Pounds of steam evaporated from water at 212° Fahr., by one pound of combustible, per INDICATOR, . | 9·188 | 10·489 |
| | Pounds of steam evaporated from water at 212° Fahr., by one pound of combustible, per TANK, . . | 12·813 | 12·445 |
| CONDENSATION. | Per centum of the steam evaporated in the boilers condensed to produce the total power developed by the engine, according to JOULE'S equivalent, . . . | 6·91 | 5·84 |
| | Per centum of the steam evaporated in the boilers not accounted for by the INDICATOR, . . | 28·29 | 15·72 |
| | Difference, due to all causes, between the weight of feed-water pumped into the boilers, according to the TANK, and the weight of steam discharged from the cylinders into the condenser at the end of the stroke of the piston, per INDICATOR, expressed in per centum of the feed-water, . . . . | 35·20 | 21·56 |

## DISCUSSION OF THE RESULTS.

The experiments recorded in the preceding table determined the following facts for the experimental conditions, namely:—

The evaporative efficiency of the boiler.

The cost of the indicated horse power both in weight of fuel and of water or steam per hour, with the steam used expansively and without expansion.

The condensation of steam in the cylinders.

And, 1st. *Of the evaporative efficiency of the boiler.* Comparing the weight of water evaporated by the pound of anthracite combustible with the same quantity given by boilers, both of other and the same type, it appears that this result from the "MONITOR'S" boiler approaches the maximum obtained from the best types, and is a maximum for its own. It becomes of value, then, to ascertain to what cause this superiority over other boilers of its own type is due. The form of the furnace and its collocation with the combustion chamber, smoke connexion, horizontal fire tubes, and uptake are the same as in other boilers of its type.

The proportion of heating to grate surface is a medium (26·52 to 1·00), but the proportion of the length of the tubes to their inner diameter is a maximum, being in the ratio of nearly 60 to 1, while the proportion of the calorimeter is a minimum, being only about one-twelfth of the grate surface. It is to the latter proportion, principally, that the high economic evaporation is due, as, by restricting the admission of air to the furnace to the minimum required for the combustion, it maintained a high furnace temperature and, consequently, proportionably reduced the per centum of the loss by the heat passing off in the products of combustion with the uptake temperature. This high furnace temperature also rendered the heating surface of maximum efficiency in consequence of the great difference between it and the temperature of the water in the boiler.

The rate of combustion was very low, the maximum being only 6 pounds of anthracite consumed per hour per square foot of grate surface, although as much was fed to the furnaces as they would consume with clean fires 7 inches thick. This low rate was not due to the small calorimeter of the tubes, but to the very small cross area of the chimney in proportion to the grate surface. This proportion was only 1·00 of chimney to 17·46 of grate, or less than $5\frac{3}{4}$ per centum of the latter.

The rate of combustion was different in the two experiments, owing to the different heights of chimney employed. In the first experiment, using the steam expansively, the chimney was 42 feet 6 inches high above the grates, and the rate of combustion was 4·519 pounds of combustible per hour per square foot of grate. In the second experiment, using the steam without expansion, the chimney was 33 feet high and the rate of combustion 3·999 pounds. These rates compare as $\left(\frac{4·519}{3·999}=\right)$ 1·000 to 1·130 : the square roots of the heights of the chimneys compare as $\left(\frac{\sqrt{42\frac{1}{2}}}{\sqrt{33}}=\right)$ 1·000 to 1·135: the equality of these proportions, it will be observed, is a practical confirmation of the theoretical law.

2d. *Of the cost of the indicated horse power.* The experiments were made using all the steam the boilers would furnish with the maximum rate of combustion by the natural draught; but this quantity was so small in comparison to the large capacity of the cylinders and little resistance, proportionally, of the screw, as to make the steam pressures in the cylinders very low. Now as the back pressure against the pistons, and the pressure required to work the engines *per se*, are constant, let the mean total pressure be what it may, it is evident that, with a low total pressure, the gross effective horse power and the net horse power will be correspondingly expensively obtained. Not so, however, the total horse power, which, as it measures the entire dynamic effect produced by the entire weight of steam used in the cylinders, whether that effect be useful or prejudicial, is independent of and unaffected by the proportion which the back pressure and pressure required to work the engines *per se*, may bear to the total pressure. For the purpose of proper comparison, then, with the economic result given by other engines, and for correct appreciation of the absolute economic result in itself, we must take, as the measure, the cost of the total indicated horse power. Whether much or little of that power is utilized depends wholly on the proportion which the sum of the back pressure and pressure required to work the engines *per se*, bears to the total pressure; and this proportion, with a given boiler pressure, depends entirely on the proportions of the organs of the engine, and of the capacity of the cylinders to the resistance momentarily opposed to the pistons. If the engines be properly designed in themselves and for their work, the proportion of the total power utilized will be a maximum and the cost of the useful work done a minimum: and just in degree as they are improperly designed, either in themselves or for their work, will the cost of the useful work done increase, although

the cost of the total power, or total work done remains the same. Therefore, in considering the cost of the power in the case of the engines of the "MONITOR," it will be taken for the total indicated horse power only. In the experimental determinations the cost of that power was ascertained in pounds of feed-water or of steam, in pounds of anthracite, and in pounds of combustible consumed per hour. Now as the two latter involve evaporative efficiency of boiler and of fuel, skill and care of firemen, and other conditions, also variable and impossible to determine or eliminate, they cannot be used for purposes of comparison as accurate measures of the cost of power by the engines *per se;* but the former is free from all objection, and where care has been taken to secure perfect tightness in all parts of the machinery, and to prevent foaming or priming in the boilers, it is critically exact, and is the true, universal, and only measure of the cost of the power. Our comparisons, then, will be made for the cost of the total indicated horse power in pounds of feed-water consumed per hour.

In all steam engines, whatever may be their type, absolute dimensions, proportions, mode of application, or nature of the resistance they overcome, the cost of the total indicated horse power in weight of steam is affected by three things only, 1st. The measure of expansion with which the steam is used; 2d. The mean total pressure of the steam in pounds per square inch on the piston during its stroke; 3d. The proportion of the steam condensed in the cylinder in addition to that which is condensed to furnish the heat transmuted into the total power developed by the engine. The latter in its turn is affected, 1st. By the measure of expansion with which the steam is used: the greater this measure the greater will be the condensation. 2d. By the extent of cylinder *interior* surface exposed in a given time to communication with the condenser, in proportion to the weight of steam used in that time; the greater this extent the greater, *ceteris paribus*, will be the condensation. This is again affected by the pressure of the steam, the temperature of the condenser being supposed constant; for the higher the steam pressure, other things equal, the greater will be the condensation. 3d. In like manner as 2d, by the extent of cylinder *exterior* surface in proportion to weight of steam used per stroke of piston; the greater this extent, the greater, *ceteris paribus*, will be the condensation. This, too, is again affected by the pressure of the steam, the temperature and condition of the atmosphere being supposed constant; for the higher the steam pressure, other things equal, the greater will be the condensation, as a natural result of a greater loss of heat by external radiation. It is here worthy of remark that the same causes which increase the condensation under the conditions of No. 2, also increase it under the conditions of No. 3.

From the description of the "MONITOR'S" engines, it will be perceived that the two cylinders occupy the same barrel, the separation being made by a simple partition of cast iron in the centre. Further, that, during a large portion of the time, the boiler steam occupies one end of one cylinder while the adjacent end of the other cylinder is open to the condenser. There is, consequently, one end of one cylinder maintained at the temperature of the boiler steam, while the adjacent end of the other cylinder, separated only by a cast iron partition, is exposed to the temperature of the condenser. This arrangement, immaterial as it appears, and is, in a *mechanical* point of view, powerfully affects the *economic* result by its great influence on the cylinder condensation. To appreciate it, it is only necessary to imagine the piston of the starboard engine, for example, to be near the outboard end of its stroke, in which case nearly the *whole* of the cylinder of that engine will be filled with steam. At this moment the piston of the port engine is near the centre of its stroke, and about one-half of the port cylinder adjacent to the starboard cylinder will be open to the condenser and exposed to its refrigerating influence; consequently, the boiler steam in the starboard cylinder has been exposed for about one-half of the stroke of its piston to this

refrigerating influence from the port cylinder, transmitted through the iron partition of the two cylinders, which, as their diameter is great in proportion to the stroke of their piston, forms a large proportion of the surface in contact with the steam. Nor does the evil end here, for as the sides of both cylinders are the same piece of iron, those of the one being merely an extension of those of the other, the conduction of heat is very rapid from one cylinder to the other, and the heat imparted by the steam to the sides of the starboard cylinder, quickly passes along, by conduction, to the sides of the port cylinder, whose interior is in communication with the condenser, and whose exterior is exposed to the atmosphere; the inevitable result, it is manifest, must be a largely increased steam condensation in cylinders of this type of engine over that in the cylinders of engines of the usual type; how much larger is a question which experiment alone can answer. There is still to be added to the already described peculiar causes of steam condensation in cylinders of the "MONITOR" type of engine, that of the half-trunk, the effect of which is, for a given capacity of cylinder, to increase both the interior and exterior cylinder surfaces, while the thin unprotected metal of the half trunk,—one side of which is always in contact with the atmosphere, while the other side is, too, for half the time, and not only in contact with but in rapid movement through it,—makes it a refrigerator of maximum power.

The experiments recorded in this volume allow the inferiority of economic effect due to this arrangement to be determined with much precision. Referring to the experiments made at Erie on the engine (of usual system of construction) of the United States paddle wheel Steamer "MICHIGAN," to determine the relative economy in rappòrt of fuel to power of using steam with different measures of expansion, it appears that when cutting off at $\frac{4}{9}$ths or 0·444 of the stroke of the piston from the commencement, the cost of the total indicated horse power was (Table No. 1, line 28) 29·416 pounds of feed-water. The cost of the same horse power in the engines of the "MONITOR" when cutting off at 0·425 of the stroke of the piston from the commencement, is 37·558 pounds, consequently in the latter it cost $\left(\frac{37·558}{29·416} = \right) 27\frac{7}{10}$ per centum more than in the former. Great as this excess appears, it is no more than what the conditions fully warrant to expect, and should be decisive against the use of such a type of engine.

In the cylinder of the "MICHIGAN'S" engine, there was 1·583 square foot of exposed interior surface per cubic foot of space displacement of piston; in the cylinder of the "MONITOR'S" engines, there was 2·792 square feet of exposed interior surface per cubic foot of space displacement of piston. In the cylinder of the "MICHIGAN'S" engine the average total pressure on the piston during its stroke was 27·1 pounds per square inch with a back pressure of 3·0 pounds per square inch; in that of the "MONITOR'S" it was 12·5 pounds, with a back pressure of 4·3 pounds.

The experiments with the "MONITOR'S" machinery give the relative cost of the total indicated horse power in pounds of feed-water per hour when using the steam expansively—cutting it off when 0·425 of the stroke of the piston was completed,—and when using it absolutely without expansion—following the piston during the entire stroke. When using the steam expansively, this cost was 37·558 pounds of feed-water per hour, and when using it without expansion it was 44·721 pounds, consequently the gain in the former case was $\left(\frac{44·721 - 37·558 \times 100}{44·721} = \right)$ 16·02 per centum of the cost in the latter case.

This determination being made for the total horse power, corresponds to the problem of doing the same work in the same time with the same cylinder, both when using the steam expansively, and when using it without expansion, consequently, in the former case the initial steam pressure in the cylinder must be carried sufficiently higher than in the latter case, to give equal average effective pressures in both on the piston

during its stroke. These are, of course, the most favorable conditions possible for the expanded steam. If, however, the load, during the experiments, could have been increased when using the steam without expansion, so as to have made the *initial* cylinder steam pressure the same as when using the steam expansively, and then the comparison have been made for the cost of the *net* indicated horse power, which is the proper mode of treating the problem, the result would have been very different, as will appear from the following considerations.

With the steam used expansively, the average total indicated pressure on the piston was 12·5 pounds per square inch, of which $(4{\cdot}3 + 1{\cdot}5 =)$ 5·8 pounds were expended in overcoming the back pressure and friction resistance of the engines *per se*, leaving $(12{\cdot}5 - 5{\cdot}8 =)$ 6·7 pounds or $\left(\frac{6{\cdot}7 \times 100}{12{\cdot}5} = \right)$ 53·60 per centum of the total pressure utilized. In this case the cylinder initial pressure was 16·6 pounds per square inch above zero. Starting with this initial pressure when using the steam without expansion, and carrying it to the end of the stroke of the piston, and having, of course, the same back pressure and friction resistance of the engines, *per se* as before, we should have for the portion utilized, $(16{\cdot}6 - 5{\cdot}8 =)$ 10·8 pounds per square inch or $\left(\frac{10{\cdot}8 \times 100}{16{\cdot}6} = \right)$ 65·06 per centum of the total pressure. Hence, when using the steam expansively the cost of the net indicated horse power would be $\left(\frac{37{\cdot}558}{0{\cdot}536} = \right)$ 70·072 pounds of feed-water per hour; and when using it without expansion $\left(\frac{44{\cdot}721}{0{\cdot}6506} = \right)$ 68·738 pounds of feed-water per hour; the cost being less in the latter case by $\left(\frac{70{\cdot}072 - 68{\cdot}738 \times 100}{70{\cdot}072} = \right)$ 1·90 per centum of the cost in the former case.

This determination, being made for the net indicated horse power and equal cylinder initial pressures, corresponds to the problem of doing the same work in the same time, but with cylinders of different dimensions; the cylinder using the steam without expansion having as much less capacity than the one using the steam expansively as the net pressure with the former is greater than the net pressure with the latter. This, of course, is the proper manner of comparing the cost of the net powers, which are the powers utilized or commercially available, and is based on the fundamental equal condition of having the *same boiler* in both cases, and, consequently, of being able to carry in it the same pressure, which gives the same initial pressure in the cylinders.

*Of the condensation of steam in the cylinders.* This, it will be seen by reference to the table of the experiments, was very large: the difference between the weight of water entering the cylinder as steam, according to the tank measurement, and present in it as steam at the end of the stroke of the piston, according to the indicator measurement, being no less than 35·20 per centum of the former, when the steam was used expansively. Of this quantity, 6·91 was due to the transmutation of heat into the power developed by the engine, and 28·29 was due to the expansion *per se* and the refrigerating influences. In the experiment on the engines of the "MICHIGAN," at Erie, already cited, the latter quantity was only 18·14 per centum. It must not, however, be supposed that this 28·29 measures the per centum of the steam which entered the cylinders that was condensed by other causes than the production of the power. The condensation was much greater, for a considerable portion of the steam present at the end of the stroke of the piston was due to re-evaporation in the cylinder by the heat imparted to its metal by the boiler steam during the first portions of the stroke of the piston, and rendered available by the reduced pressure in the cylinder during the last portions of the stroke of the piston.

When the steam was used without expansion, the difference between the weight of water which entered the cylinder as steam, and the weight which was present in it as steam at the end of the stroke of the piston, was 21·56 per centum of the former. Of this 5·84 was due to the transmutation of heat into the power developed by the engines, and 15·72 was caused wholly by the refrigerating influences. The 15·72 may be considered in this case as the whole condensation due to other causes than the production of the power, for the difference in the cylinder steam pressures at the commencement and at the end of the stroke of the piston being only 1·9 pound per square inch, there could have been scarcely any re-evaporation.

# EXPERIMENTS

MADE WITH THE

MACHINERY OF THE U. S. IRON-CLAD STEAM BATTERY

# "PASSAIC."

# EXPERIMENTS

MADE WITH THE

# MACHINERY OF THE U. S. IRON-CLAD STEAM BATTERY "PASSAIC,"

TO DETERMINE THE COST OF THE INDICATED HORSE POWER IN POUNDS OF STEAM AND OF FUEL PER HOUR WHEN USING THE STEAM WITHOUT EXPANSION; AND TO DETERMINE THE EVAPORATIVE EFFICIENCY OF THE BOILER WITH ANTHRACITE.

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ADVANTAGE was taken of the opportunity afforded by the making of some repairs on the machinery of the U. S. iron-clad steam battery "PASSAIC," at the Washington Navy Yard, D. C., to experiment with it, after it had been put in perfect order, with a view to ascertain the cost of the indicated horse power as given by it, both in weight of fuel and of steam, and the economic evaporative efficiency of the boiler when burning anthracite.

In making these experiments, it was the intention to have repeated with the machinery of this vessel those that were made with the machinery of the U. S. iron-clad steam battery "MONITOR," described in the immediately preceding paper, but the necessity for the services of the vessel interrupted them when only the one in which the steam was used without expansion had been finished, leaving the cost of the indicated horse power undetermined when using the steam expansively.

The machinery was in perfect order, every part having been carefully examined. The cylinder, piston, and valves had been overhauled, and made tight, and the engine had been re-lined. The boiler, when tested with a hydrostatic pressure of 40 pounds per square inch above the atmosphere, was found perfectly tight also. It was quite new and thoroughly covered with felt.

The ventilation of the fire room and the combustion of the fuel in this vessel, depend on fan-blowers worked by small auxiliary engines. The fire room is air-tight, and the air is driven into it, and thence into the ash-pits, and out of the smoke-pipe. During the experiments, however, the blowers were not used, and the anthracite was burned with the natural draught, the air for which entered through the open fire room hatches.

The following is a description of the machinery, namely:—

## ENGINES.

The engines are the "vibrating lever" kind, patented by JOHN ERICSSON, and are in all respects—to the minutest detail—duplicates of those of the "MONITOR," a description of which will be found under the title of that vessel in the immediately preceding paper.

## SCREW.

There is one cast iron true screw. The front and back edges of the blades are straight and parallel, and slope backwards 5½ inches in their radial length from hub to periphery.

| | |
|---|---|
| Diameter of the screw, | 12 feet. |
| Diameter of the hub at centre of its length, | 1 foot 8 inches. |
| Diameter of the hub at its ends, | 1 " 5½ " |
| Pitch, | 16 feet. |
| Number of blades, | 4. |
| Length of the blades in the direction of the axis of the screw, | 2 feet. |
| Fraction used of the pitch, | 0 5 |
| Helicoidal surface of the four blades, | 68·10 square feet. |
| Projected surface of the four blades on a plane at right angles to axis of the screw, | 55 65 " |
| Thickness of the blades above the fillet at the hub, | 6 inches. |
| Thickness of the blades at the periphery, | 1 inch. |
| Weight of the screw, | 8,140 pounds. |

## BOILERS.

The boilers are two in number, and are placed side by side with a space of 8 inches between them. They are of the vertical water-tube type, with the tubes arranged above the furnaces, according to the patent of D. B. MARTIN. There is one chimney in common for both furnaces.

Each boiler contains three furnaces: each furnace is semicircular on top, is 33 inches wide, 6 feet 11 inches long, and has a length of grate-bars of 6½ feet. The height from the top of the furnace to the bottom of the ash-pit is 38 inches. The angles of the ash-pits are rounded with quadrantal arcs of 10 inches radius. The water-bottoms beneath the ash-pits are 4 inches wide, including thicknesses of metal; the water-legs between the furnaces are 5 inches wide, including thicknesses of metal; and the water-legs between the furnaces and the shell of the boiler are 5½ inches wide, including thicknesses of metal. The water space at the front of the furnace is 5 inches wide, including thicknesses of metal. The height from the crown of the furnace to the top of the grate-bars at front of furnace is 20 inches, and at back of furnace 26 inches.

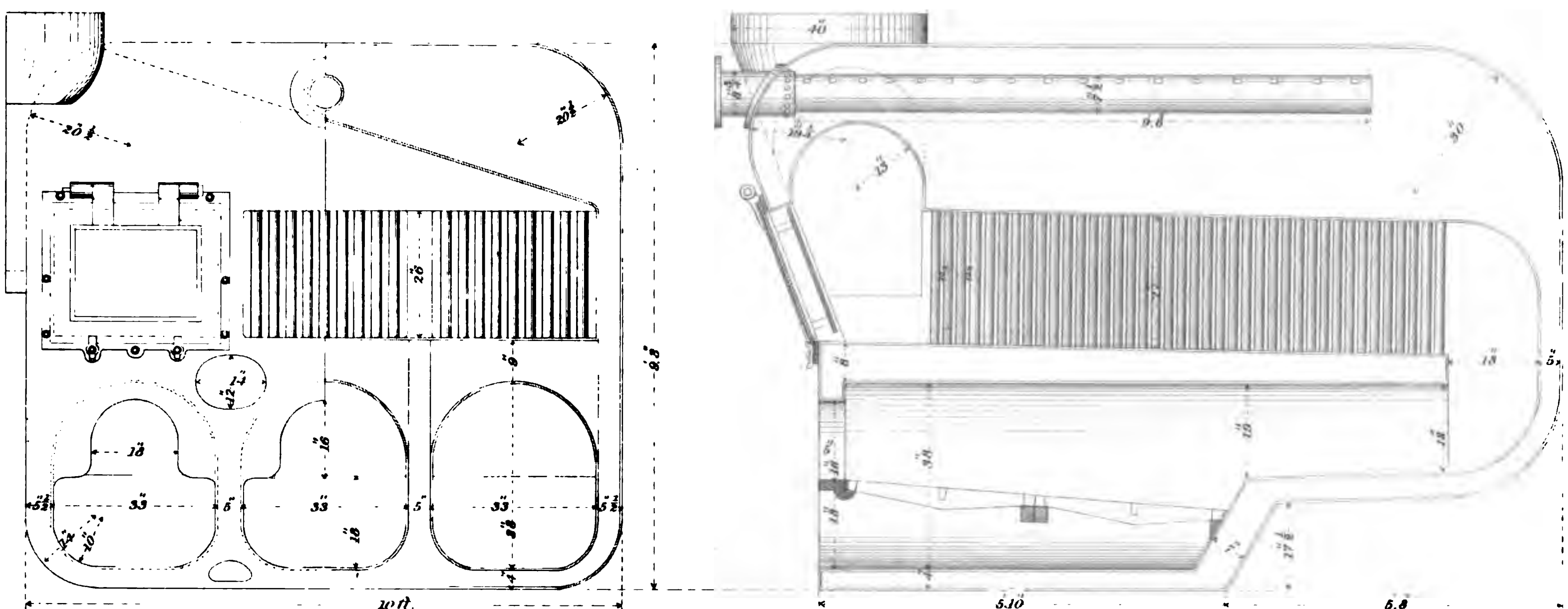
The opening for the furnace door is semicircular on top, 16 inches high, and 18 inches wide in the clear. The door is of cast iron, and both it and its lining plate are pierced with four openings, each of 4½ square inches area, making a total area of 18 square inches for each furnace.

PLATE XVII.

## BOILER FOR U. S. STEAMER

## PASSAIC

Scale 1/12 inch - 1 foot




Half Elevation — Half Section on A.B.

Longitudinal Section through Furnace, Tube box, Uptake, and Shell.

From each furnace a combustion chamber extends horizontally 39 inches to the back smoke connexion. The top of this chamber is an extension of the semi-cylindrical top of the furnace. The extreme height of the chamber at the furnace end is 19 inches, and at the smoke connexion end 18 inches. The bottom is flat and is connected to the sides by arcs of 3 inches radii. The water-bottom beneath the chamber is 5 inches wide, including thicknesses of metal.

There is a separate back smoke connexion for each furnace. It is 18 inches wide, and 4 feet 3 inches in extreme height. The sides and back are flat, and the top and bottom are composed of quadrantal arcs of 18 inches radii. The water space between back of connexion and boiler shell is 5 inches wide, including thicknesses of metal.

There is a separate tube-box for each furnace, lying immediately over it and the combustion chamber. It is 33 inches wide and 26 inches high in the clear. Total length 8 feet 10 inches. The bottom tube-plate at front of furnace is 9 inches above furnace crown, and at back of combustion chamber it is 6 inches above crown of that chamber, including thicknesses of metal. The water spaces between tube-boxes are 5 inches wide, and between tube-boxes and boiler shell $5\frac{1}{2}$ inches wide, including thicknesses of metal.

Each tube-box contains three hundred and sixty lap-welded iron tubes, distributed into nine rows crosswise the box, and forty rows lengthwise the box. The tubes in the lengthwise rows have not their centres in the same straight line, but in two straight lines, alternately; these lines are parallel and $\frac{3}{16}$ths of an inch apart. By this arrangement the tubes are zigzaged or staggered $\frac{3}{16}$ths of an inch in the lengthwise direction of the tube-box. Each tube is 2 inches in external diameter and 26 inches loug in the clear between the tube-plates, to which they are secured by being expanded ou the one side and riveted over on the other.

The uptake is at the front of the boiler above the furnaces, and is in common for all of them. It is 21 inches wide, and has its front closed by three cast iron doors with wrought iron lining plates. These doors close against two water-legs of 5 inches square in section, placed in front of the uptake, and immediately over the water spaces between the furnaces.

There is a man-hole of 14 inches by 12 inches between the furnaces, and in the steam room of the boiler. Also hand-holes between the ash-pits.

The shell of the boiler is composed of $\frac{3}{8}$-inch thick plate, with the exception of the ash-pit bottoms, which are of $\frac{7}{16}$-inch thick plate. The bottoms of the ash-pits are also of $\frac{7}{16}$-inch thick plate. The tube-plates are $\frac{1}{2}$-inch thick. The metal of the tubes is $\frac{1}{12}$-inch thick. All other parts are of $\frac{3}{8}$-inch thick plate. All seams not in contact with the products of combustion are double riveted. In addition to the regular bracing, the top and sides of the shell are stiffened by T-iron placed every 12 inches, and extending around from the level of the grates to the same level on the opposite side.

The following are the principal dimensions and proportions of the boilers, namely:—

| | |
|---|---|
| Length of each boiler at top, | 13 feet 7 inches. |
| Length of each boiler at top of furnaces, | 12 " 6 " |
| Breadth of each boiler, | 10 " |
| Height of each boiler, | 9 " 3 inches. |
| Number of furnaces in the two boilers, | 6. |
| Breadth of each furnace, | 2 feet 9 inches. |
| Length of grate-bars, | 6 " 6 " |

| | |
|---|---|
| Area of grate surface, | 107·25 square feet. |
| Number of tubes in the two boilers, | 2,160· |
| External diameter of tubes, | 2 inches. |
| Internal diameter of tubes, | $1\frac{3}{8}$ inch. |
| Length of tubes between tube-plates, | 2 feet 2 inches. |
| Heating surface in the six furnaces, | 252 square feet. |
| Heating surface in the six combustion chambers, | 144 " |
| Heating surface in the six back smoke connexions, | 181 " |
| Heating surface in the six tube-boxes, | 431 " |
| Heating surface in the tubes of both boilers, | 2,450 " |
| Heating surface in the uptakes of both boilers, up to 9 inches above top of tube-plate, | 70 " |
| Total water heating surface in the two boilers, | 3,528 " |
| Total steam superheating surface in the uptakes of the two boilers, | 35 " |
| Length of space occupied by the tubes from forward side of front tube to after side of back tube, | 8 feet 6 inches. |
| Distance in the clear between the tubes, lengthwise the boiler, | 0·564 inch. |
| Distance in the clear between the tubes, crosswise the boiler, | 1·500 " |
| Diameter of the chimney, | 3 feet 6 inches. |
| Height of chimney above level of grates, | 27 " |
| Aggregate cross area of the combustion chambers at back, | 19·734 square feet. |
| Aggregate absolute cross area between the tubes, | 16·250 " |
| Aggregate cross area between tubes for direct draught, | 13·325 " |
| Cross area of the chimney, | 8·726 " |

## PROPORTIONS.

| | |
|---|---|
| Ratio of the water heating to the grate surface, | 32·895 to 1·000 |
| Ratio of the grate surface to the cross area of the combustion chambers at their back, | 5·435 " |
| Ratio of the grate surface to the absolute cross area between the tubes, | 6·600 " |
| Ratio of the grate surface to the cross area between the tubes for direct draught, | 8·049 " |
| Ratio of the grate surface to the cross area of the chimney, | 12·291 " |
| Ratio of the steam superheating surface to the grate surface, | 0·326 " |
| Steam room in the two boilers, | 485· cubic feet. |
| Weight of water in the two boilers, | 28,635· pounds. |
| Weight of the two boilers, including all doors and plates, but excluding grate bars and chimney, | 73,400· " |

## MANNER OF MAKING THE EXPERIMENTS.

During the making of the experiments the vessel was secured to the wharf of the Washington Navy Yard, D. C., and the power developed by the engines was employed in forcing the water astern by the screw propeller.

The power developed by the engines was ascertained by two indicators, one permanently attached to each cylinder and not removed during the experiments; and by a self-registering counter worked by the engines. The indicators were placed near the centre of a pipe connecting the two ends of the cylinder; and, by means of suitable cocks, could be placed in communication with either end. They were excellent instruments, graduated to 12 pounds pressure per square inch per inch, and gave true and smooth diagrams.

The cost of the power was ascertained in pounds of anthracite, pounds of combustible, and pounds of feed-water or steam consumed per hour.

The anthracite consumed, and the refuse from it in ashes, clinker, dust, &c., were accurately weighed on a new Fairbanks platform scales; the refuse in a dry state.

The feed-water, before being pumped into the boiler, was measured with scrupulous care in a wooden tank lined with sheet lead, which discharged at each delivery 55·605 cubic feet of water into the boiler. The tank was placed upon the wharf with its bottom, at all states of the tide, above the deck of the vessel.

It was filled from the Navy Yard reservoir by gravity through a hose with a pipe and stop-cock at the end. As soon as the tank was filled this hose was thrown out to insure against possible leakage into the tank while it was discharging into the boiler. From the bottom of the tank to the receiving nozzle of the engines feed-pump there was connected a leather hose with a stop-cock close to the bottom of the tank. The suction-pipe between the feed-pump and hot-well had been removed to permit this arrangement, and the aperture in the hot-well closed by a blank flange. The tank was mounted on supports which allowed it to be viewed all around and beneath. The hose connecting it with the feed-pump was in full view for its whole length ; and the feed-pipe from the pump to the boiler, being overhead, was also in full view for its whole length. There was no possibility of the leakage of a drop of water without discovery. The tightness of the boilers was tested by allowing the water to stand in them under the average steam pressure for a number of hours, and no appreciable fall could be observed. It is believed that the entire apparatus of tank, pump, hose, pipes, and boilers were perfectly tight, and that there was no leakage of water from any cause out of them.

The blow-off valves of the boilers had just been ground in tight,—as had indeed all the cocks and valves of the machinery, when the vessel was out of water on the ways,—they were carefully secured closed, and were not opened during the experiments.

The boilers were provided with glass water gauges of the usual construction. The temperature of the injection-water was obtained from a thermometer immersed overboard; and of the feed-water from a thermometer kept constantly immersed in the tank. The temperature of the hot-well was ascertained by drawing water from it upon a thermometer immersed in a pot. The steam pressure in the boilers, and the vacuum in the condenser, were denoted by open mercurial syphon gauges of the usual construction. The height of the barometer was taken from an instrument of the usual construction for the Navy in a neighboring vessel.

The experiments were conducted by the engineer department of the vessel. The firemen were expe-

rienced, and all were familiar with the machinery. The Assistant Engineers were in four watches of four hours each, and personally attended to the tank, the weighing, the taking of the indicator diagrams, and the keeping of the record, &c. The firemen of each watch were regularly posted; each had his special service and nothing more to perform.

Just previous to the experiments the machinery had been entirely overhauled, new brasses put in, new joints made, all cocks and valves ground in, the cylinders' steam and cut-off valves re-faced and scraped to their seats, and the packing of the pistons carefully set up and re-adjusted. It is believed that both valves and pistons were perfectly tight, and that there was no leakage of steam past them.

The experiment lasted uninterruptedly fifty-five hours and ten minutes, during which the steam was used absolutely without expansion, for the steam slide-valve had neither lap nor lead. The independent slide cut-off valve was working as usual, and steam was given during the whole stroke of the piston by keeping the pass-over valve open.

There was one screw throttle-valve in the main steam-pipe for both engines, and it was kept at the same opening throughout, being nearly closed. The boiler pressure varied but slightly, and care was taken that the conditions should remain as uniform as possible. The steam pressure, vacuum, temperatures, speed of piston, rate of combustion of the fuel, &c., varied between only very trifling limits. The water in the boiler was carried at 6 inches above the tubes, and was fed in very uniformly by means of the cock in the suction hose at the bottom of the tank. The fires were carried 7 inches thick upon the grates, and were kept well cleaned, leveled, and free from holes. The anthracite was burnt with the natural draught, and, although the fires were not forced, yet as much was burned as they would take; the rate of combustion was, therefore, the maximum under the conditions.

Before commencing each experiment, the machinery was operated about 6 hours to adjust it to the normal conditions required to be uniformly maintained during its continuance, and to bring the fires to steady action. No note was taken of the water and coal thus expended. When all was ready, average fires and steam pressure being in the boilers, and the water in the glass gauge being at the 6 inches above the tubes, the tank was filled, the time and the number on the counter noted, and the experiment commenced. From this time to its end all the quantities were systematically weighed or measured, and noted. At the end of the experiment, the water in the boilers was brought to exactly the same level as at the beginning; and the fires, having been thoroughly cleaned, were made the same thickness as at the beginning.

In conducting the experiments a tabular record was kept in which was entered at the end of each hour in appropriate columns, the number on the counter; the average number of double strokes of engines' pistons made per minute during that hour; the average steam pressure in the boilers; the average vacuum in the condenser; the temperature of the injection-water; of the hot-well; of the feed-water; of the engine room; and of the external atmosphere in the shade; and the number of pounds of anthracite thrown into the furnaces during that hour. The weight of refuse, and the height of the barometer were noted in their respective columns at the close of each watch of four hours. In another column was recorded the exact time at which each tankful of water was emptied. At the end of every hour an indicator diagram was taken from each end of each cylinder.

The data and results of the experiments made with the machinery, in the manner and under the conditions hereinbefore described, will be found in the following table, namely:—

## TABLE CONTAINING THE DATA AND RESULTS OF THE EXPERIMENT MADE WITH THE MACHINERY OF THE U. S. IRON-CLAD STEAM BATTERY "PASSAIC" AT THE WHARF OF THE WASHINGTON NAVY YARD, D. C., TO DETERMINE THE COST OF THE INDICATED HORSE POWER IN POUNDS OF STEAM AND OF COAL PER HOUR, WHEN USING THE STEAM WITHOUT EXPANSION; AND THE EVAPORATIVE EFFICIENCY OF THE BOILERS WITH ANTHRACITE.

### USING THE STEAM WITHOUT EXPANSION.

Date of commencing the experiment, . . . . . Meridian, Dec. 15, 1862.

State of the weather, . . . . . Half time moderate wind with rain. Half time moderate wind and cloudy.

| | | |
|---|---|---|
| TOTAL QUANTITIES. | Duration of the experiment, in consecutive hours, | 55⅛ |
| | Number of double strokes made by the engines' pistons, per counter, | 92,100· |
| | Pounds of feed-water pumped into boilers, per tank, | 301,142· |
| | Pounds of anthracite consumed, | 32,022· |
| | Pounds of refuse from the anthracite, in ashes, clinker, soot, and dust, | 5,088· |
| | Pounds of combustible consumed, | 26,934· |
| | Per centum of refuse from the anthracite, | 15·89 |
| QUANTITIES PER HOUR. | Pounds of feed-water pumped into boilers, | 5,458·766 |
| | Pounds of anthracite consumed, | 580·459 |
| | Pounds of combustible consumed, | 488·230 |
| | Pounds of anthracite consumed per square foot of grate, | 5·412 |
| | Pounds of combustible consumed per square foot of grate, | 4·552 |
| ENGINES. | Number of double strokes made per minute by the engines' pistons, | 27·825 |
| | Vacuum in condenser, in inches of mercury per gauge, | 22·0 |
| | Height of barometer, in inches of mercury, | 29·9 |
| | Fraction of the stroke of piston completed when the steam was cut off, | 1· |
| | Steam pressure in boilers, in pounds per square inch above atmosphere, per gauge, | 27·8 |
| | Area of the throttle-valve open, in square inches, | 8· |
| STEAM PRESSURE IN CYLINDERS, PER INDICATOR. | In pounds per square inch above zero at commencement of stroke of piston, | 18·10 |
| | In pounds per square inch above zero at end of stroke of piston, | 15·00 |
| | In pounds per square inch above zero against the piston during its stroke, | 6·76 |
| | Mean gross effective pressure in pounds per square inch of piston during its stroke, | 9·74 |
| | Mean total pressure in pounds per square inch of piston during its stroke, | 16·50 |
| | Mean net pressure in pounds per square inch of piston during its stroke, | 8·24 |

| | | | |
|---|---|---|---|
| POWER. | ABSOLUTE. | Gross effective horses power developed by the engines, | 71·372 |
| | | Total horses power developed by the engines, | 120·907 |
| | | Net horses power developed by the engines, | 60·380 |
| | ECONOMIC. | Pounds of feed-water consumed per hour per gross effective horse power, | 76·483 |
| | | Pounds of feed-water consumed per hour per total horse power, | 45·148 |
| | | Pounds of feed-water consumed per hour per net horse power, | 99·407 |
| | | Pounds of anthracite consumed per hour per gross effective horse power, | 8·133 |
| | | Pounds of anthracite consumed per hour per total horse power, | 4·809 |
| | | Pounds of anthracite consumed per hour per net horse power, | 9·613 |
| | | Pounds of combustible consumed per hour per gross effective horse power, | 6·841 |
| | | Pounds of combustible consumed per hour per total horse power, | 4·038 |
| | | Pounds of combustible consumed per hour per net horse power, | 8·086 |
| TEMPERATURE | IN DEGREES FAHR. | Of the injection-water, | 87·6 |
| | | Of the hot-well, | 100·0 |
| | | Of the feed-water, | 46·6 |
| | | Of the engine room, | 96·7 |
| | | Of the external atmosphere in the shade, | 49·7 |
| TOTAL EVAPORATION. | FROM TEMP. OF FEED-WATER | Pounds of steam discharged from cylinders into condenser, calculated from the pressure of the steam at the end of the stroke of the piston, | 235,951·029 |
| | | Pounds of steam equivalent to the heat annihilated in the cylinders to produce the total power of the engines, calculated from JOULE'S equivalent, | 17,724·108 |
| | | Pounds of steam evaporated from water at 100° Fahr., per INDICATOR, | 266,240·068 |
| | | Pounds of steam evaporated from water at 100° Fahr., per TANK, | 316,057·983 |
| | | Pounds of steam evaporated from water at 212° Fahr., per INDICATOR, | 297,105·287 |
| | | Pounds of steam evaporated from water at 212° Fahr., per TANK, | 352,698·594 |

| | | |
|---|---|---|
| ECONOMIC EVAPORATION. | Pounds of steam evaporated from water at 100° Fahr., by one pound of anthracite, per INDICATOR, | 8·314 |
| | Pounds of steam evaporated from water at 100° Fahr., by one pound of anthracite, per TANK, | 9·870 |
| | Pounds of steam evaporated from water at 212° Fahr., by one pound of anthracite, per INDICATOR, | 9·278 |
| | Pounds of steam evaporated from water at 212° Fahr., by one pound of anthracite, per TANK, | 11·014 |
| | Pounds of steam evaporated from water at 100° Fahr., by one pound of combustible, per INDICATOR, | 9·885 |
| | Pounds of steam evaporated from water at 100° Fahr., by one pound of combustible, per TANK, | 11·734 |
| | Pounds of steam evaporated from water at 212° Fahr., by one pound of combustible, per INDICATOR, | 11·031 |
| | Pounds of steam evaporated from water at 212° Fahr., by one pound of combustible, per TANK, | 13·095 |
| CONDENSATION. | Per centum of the steam evaporated in the boilers condensed in the cylinders to produce the total power developed by the engines, according to JOULE's equivalent, | 5·88 |
| | Per centum of the steam evaporated in the boilers not accounted for by the INDICATOR, | 15·76 |
| | Difference, due to all causes, between the weight of feed-water pumped into the boilers, according to the TANK, and the weight of steam discharged from the cylinders into the condenser at the end of the stroke of the piston, per INDICATOR, expressed in per centum of the feed-water, | 21·64 |

## DISCUSSION OF THE RESULTS.

The experiments recorded in the preceding table determined the following facts for the experimental conditions, namely:—

The evaporative efficiency of the boiler.

The cost per hour of the indicated horse power when using the steam absolutely without expansion, both in weight of fuel and of steam or water.

The condensation of steam in the cylinders.

And 1st. *Of the evaporative efficiency of the boiler.* If we observe the weight of water evaporated per pound of combustible, (11·734 pounds of water from the temperature of 100° Fahr.) we will see that the economic evaporation reached the maximum. Comparing it with the economic evaporation (mean of the two experiments $\frac{11{\cdot}497+11{\cdot}168}{2}=11{\cdot}332$ pounds of water from the temperature of 100° Fahr.) given by the horizontal tubular boiler of the "MONITOR," and which was obtained with the best possible proportions for that type of boiler, and with the very slow combustion of $\left(\frac{4{\cdot}519+3{\cdot}999}{2}=\right)$ 4·259 pounds of combustible per square foot of grate surface per hour, a most favorable condition with the horizontal tubes, we find the vertical water tubes to be $\left(\frac{11{\cdot}734-11{\cdot}332\times 100}{11{\cdot}332}=\right)$ $5\frac{1}{2}$ per centum superior. The rate of combustion with the vertical water tube boiler was nearly the same as with the horizontal tubular boiler, namely, 4·552 pounds of combustible per hour; but there is this difference in the two types of boiler, that the vertical water tube will give the same high economic evaporation with its maximum combustion of $12\frac{1}{2}$ pounds of anthracite per hour per square foot of grate surface, while the horizontal tubular rapidly falls in its economic evaporation as its rate of combustion is increased, until when reaching $13\frac{1}{2}$ pounds of anthracite per square foot of grate surface, there is an inferiority of from one-seventh to one-fifth, according to proportions of calorimeter and heating surface. In the case of the experiments with the boilers of the "MONITOR" and of the "PASSAIC," the combustion was so slow as to prevent any just comparison of the efficiency of the two types under their actual proportions, but they are valuable and interesting as showing the economic results possible with such rates of combustion.

2d. *Of the cost of the indicated horse power.* The main object of the experiment was to ascertain the cost of the total indicated horse power in pounds of steam per hour, as given by engines of this type. Had not the experiment been interrupted, the determination would have been made for the power both when using the steam with and without expansion; as it was, the cost was ascertained for the latter case alone.

The engines being the precise duplicates—to the minutest detail—of those of the "MONITOR," with which the same experiments were made, (see the immediately preceding paper) the experiment with the engines of the "PASSAIC," become valuable as confirmatory of their accuracy. When so peculiar a result as the excessive cylinder condensation given by engines of this type is observed, it is of the first importance to place it beyond doubt, to do which it is necessary to repeat the experiment for the purpose of assuring that the result is not due to accidental circumstances, but is inseparable from the type of engine.

When using the steam without expansion, the cost of the total horse power with the engines of the "MONITOR" was 44·721 pounds of steam per hour, and with the engines of the "PASSAIC" 45·148 pounds.

The total indicated horses power developed in the first case was 90·573, and in the latter case 120·907. The remarkably close agreement of the cost of the power in the two cases, is a proof both of the accuracy of the experiments, and that the result is due to and inseparably connected with this type of engine. It is at least one-eighth higher than in engines of the usual type. The reasons why the "MONITOR" arrangement of engines is so much more expensive in fuel, will be found discussed at length in the immediately preceding paper.

3d. *Of the condensation of steam in the cylinder.* Comparing the results of the experiment with the engines of the "PASSAIC" when using the steam without condensation, with those from the engines of the "MONITOR" under the same conditions, we find the cylinder condensations to be almost identical. In the former, the per centum of steam generated in the boiler which was condensed in the cylinder to furnish the heat transmuted into the total power of the engine, is 5·88; in the latter it is 5·84. In the former the per centum of steam generated in the boiler which was condensed in the cylinder in addition to the preceding, is 15·76; in the latter it is 15·72; proving the accuracy of the experiments, the excellence of the instruments, and the generality of the facts.

THE END.